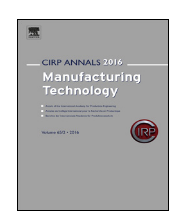
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Chatter suppression techniques in metal cutting



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ABSTRACT

The self-excited vibration, called chatter, is one of the main limitations in metal removal processes. Chatter may spoil the surface of the part and can also cause large reduction in the life of the different components of the machine tool including the cutting tool itself. During the last 60 years, several techniques have been proposed to suppress chatter. This keynote paper presents a critical review of the different chatter suppression techniques. Process solutions with design and control approaches are compiled to provide a complete view of the available methods to stabilize the cutting process. The evolution of each technique is described remarking the most important milestones in research and the corresponding industrial application. The selection of the most appropriate technique for each specific chatter problem is also discussed considering various aspects of machining processes.

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1. Introduction

Chatter in machining is a classic problem that limits the productivity. As early as 1907, Frederick Taylor stated that "chatter is the most obscure and delicate of all problems facing the machinist, and in the case of castings and forgings of miscellaneous shapes probably no rules or formulas can be devised which will accurately guide the machinist" [300].

The appearance of chatter on machine tools is disastrous since they prevent from obtaining the required surface finishes and decrease the life of tools and mechanical components. These vibration occurs in a wide range of machining operations (Fig. 1), and it is still one of the major limitations for productivity.

The recent advances in industry, especially aerospace, mould and automotive sectors, have encouraged a considerable evolution in machine tools, which became more powerful, precise, rigid and automatic. However, new limitations and challenges also showed up such as machine vibrations. After the first observations of Taylor [300], the regenerative effect was reported as the main reason of chatter by Tlusty and Polacek [310] and Tobias and Fishwick [313]. Since then, the suppression of these self-excited vibrations has become one of the major concerns and the current situation indicates that the prediction and suppression of chatter will remain as an essential problem also in the future. The main reasons are summarized below:

- *Increasing material removal rate (MRR)*

  The evolution in material technology allows increasing the cutting conditions and material removal rate (MRR). Developed around 1900, HSS tools cut four times faster than the carbon steels they replaced. Nowadays, carbide tools, which have replaced HSS tools in most applications, can cut about 3–5 times faster than HSS tools [257]. This higher capacity in

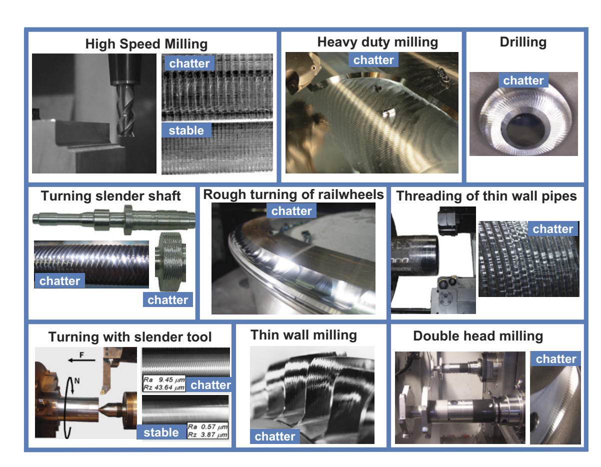


Fig. 1. Chatter problem in numerous machining applications.

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conjunction with the increase in rated power of new generation machines increases the risk of machining vibrations onset.

- *Limitations in the design procedure*

  Finite Element Method (FEM) can provide dynamic properties like natural frequencies and mode shapes in the design phase with reasonable accuracy. However, it is difficult to estimate damping which is one of the most important dynamic parameters to predict stability. This difficulty comes from the fact that joints are the main agents dissipating vibration energy, and their damping behaviour cannot be accurately predicted so far [52,163,251]. Consequently, machine designers mainly focus on increasing the static stiffness; and the result can lead to a rigid but poorly damped machine tool.

- *Low friction guiding systems*

  Rising needs in terms of accuracy have brought along an evolution in guiding systems. The first machine tools were guided through frictional guiding systems which provide relatively high damping via friction. In order to increase precision and speeds, roller bearing or aerostatic guiding systems are introduced. These guiding systems are weakly damped, and therefore, they also jeopardize machining stability.

- *Light weight design*

  Eco-efficiency is an increasing concern among machine tool builders. Machine tools manufacturers face the challenge of conceiving machines that are capable of maintaining the productivity, while consuming at the same time the least possible amount of material and energy [357]. The competition in achieving larger accelerations also leads to lighter machines which are more prone to the appearance of vibrations.

- *Manufacturing of flexible parts*

  Manufactured parts have also become lighter and less stiff, in order to minimize costs or fuel consumption in transports. Aerospace industry is the best example, where the parts must be as light as possible. The MRR requirement, together with the thin walls, makes these parts an important source of chatter (Fig. 2).

Due to the listed factors, chatter is and will be a crucial problem for the metal cutting industry.

Chatter, as a kind of self-excited vibration, depends on many factors, such as dynamic stiffness of the structure and/or the tool, cutting parameters, workpiece and tool characteristics. Mathematically, the most general case is described by a delay differential equation (DDE) with time dependent coefficients:

$$\mathbf{M}\ddot{\mathbf{r}}(t) + \mathbf{C}\dot{\mathbf{r}}(t) + \mathbf{K}\mathbf{r}(t) = \mathbf{F}_s(t) + K_t a\mathbf{A}(t)(\mathbf{r}(t) - \mathbf{r}(t-\tau)) + \mathbf{F}_{pd}(t, \dot{\mathbf{r}}(t)). \quad (1)$$

Here, $\mathbf{r}(t)$ denotes the displacement vector in Cartesian coordinates, while $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ stand for the system mass, damping and stiffness matrices. Thus, the left-hand-side presents a multi-degree-of-freedom damped oscillator that models the system formed by the tool, tool holder, spindle, machine tool structure, fixture and workpiece. The cutting force on the right-hand-side is typically formed by three terms: the time periodic stationary part $\mathbf{F}_s$ causing forced vibrations, the second term is the dynamic part related to the regenerative effect, and the third term is the process damping force $\mathbf{F}_{pd}$. In addition, the dynamic force term involves further parameters like the cutting force coefficient K_t, the depth of cut a, the regenerative delay τ and the Cartesian directional matrix $\mathbf{A}(t)$,



Fig. 2. Examples of machining of titanium impellers.

which includes the projection of the vibration onto the chip direction and the projection of the cutting force onto the Cartesian directions.

All these factors give rise to a complex problem but at the same time it allows tackling the problem from various perspectives according the different terms of Eq. (1):

- *Process parameter selection (a, τ)*

  Chatter problems can be avoided by selecting process parameter by means of Stability Lobe Diagrams (SLD) (Section 3).

- *Regeneration disturbance (τ)*

  The regeneration can be reduced by varying the delay with the help of special tool geometries (Section 4) or spindle speed variation techniques (Section 7).

- *Process damping maximization ($\mathbf{F}_{pd}$)*

  Process damping can be increased using special edge geometries (Section 4).

- *System stiffness enhancement ($\mathbf{K}$)*

  Stiffness can be increased by different procedures (Section 5).

- *System damping enhancement ($\mathbf{C}$)*

  The damping of the system can be increased using passive (Section 6) or active (Section 8) techniques.

2. Concepts for chatter suppression technique selection

One of the main goals is to define the best chatter suppression technique for each chatter case. The most suitable technique should be selected by considering different aspects of the chatter problems which are classified by the following criteria:

2.1. Machinability

Several chatter suppression methods are based on the variation of the cutting conditions, especially the cutting speed. Therefore, the machinability is an important factor when selecting the best chatter suppression technique. Materials with good machinability permit changes in the spindle speed to avoid chatter. With low machinability, however, the range of spindle speed is limited, and the objective is to move the most stable zone of SLD to the best machining conditions [16].

2.2. Relative location in the stability diagram

The qualitative location of the chatter process on the stability diagram is a key factor to select the optimal chatter suppression technique. The relative position of the unstable process is defined by the ratio k between the chatter frequency f_c and tooth passing frequency f_z which depends on the spindle speed N and the number of teeth Z on the tool (Eq. (2)). Physically, it defines the number of complete waves per period produced by the chatter. It is possible to identify four relative zones according to this ratio [282] (Fig. 3).

$$k = \frac{f_c}{f_z} = \frac{60 f_c}{Z N} \ (\text{with } N \text{ in rpm}). \quad (2)$$

- *Zone A: process damping zone ($k > 10$)*

  Process damping is important in this zone, and therefore a high increase in the stability is obtained due to the friction between the flank face with the wavy surface. In this zone, the lower the spindle speed is, the higher the stability boundary is.

- *Zone B: intermediate zone ($10 > k > 3$)*

  The stability limit is close to the absolute stability limit in the whole spindle speed range. This is especially true for high damping values.

- *Zone C: high speed zone ($3 > k > 0.5$)*

  In this zone, the stability can be drastically increased by means of the selection of the spindle speed coincident with one of the stability pockets.

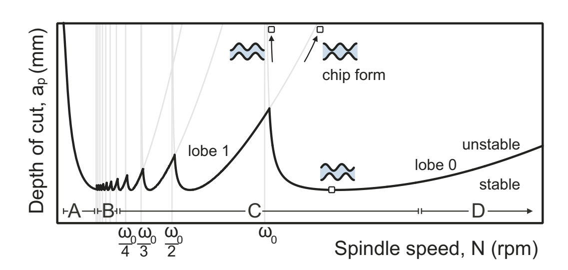


Fig. 3. Relative location on the SLD.

- *Zone D: ultra-high speed zone* (0.5 > *k*)

The stability can be improved by increasing the spindle speed. The limits of machinability and the power of the spindle in conjunction with the presence of modes at higher frequencies usually limit this otherwise promising option.

It should be noticed that each mode has a spindle speed range where it can create chatter. If the tooth passing frequency is low related to the natural frequency of the mode, the process damping may stabilize the cutting process. On the other hand, if the tooth passing frequency is several times higher than the natural frequency, the mode can hardly create chatter problems.

2.3. Critical element

Chatter grows when one or more vibration modes of the machining system are self-excited. The definition of the shape of these critical modes is one of the key factors for an efficient resolution of chatter. The dynamic flexibility of any part of the system including the machine tool structure, spindle, tool/toolholder, the fixturing system or the part to be machined can produce chatter (Fig. 4).

The chatter problems can be caused by large mass and low frequency (20–200 Hz) modes related to machine tool structure [216], or high frequency (0.5–10 kHz) local modes associated with the flexibility of the spindle and the tool [20]. The dynamic properties can change due to variations in the position of the machine and the orientation of the tool [213] (Fig. 4) or due to the material removal process [70].

The main chatter suppression techniques are reviewed in the following sections.

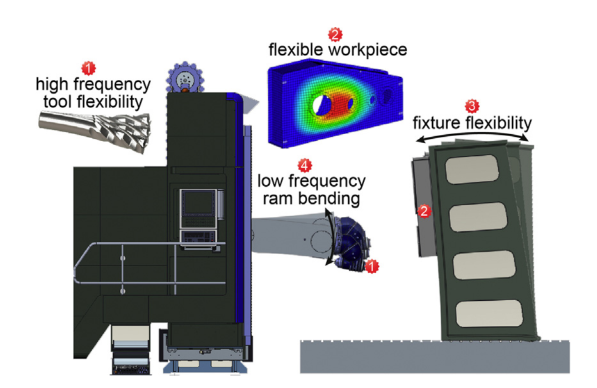


Fig. 4. Possible critical modes in a ram type milling machine.

3. Stable process planning using stability lobe diagram (SLD)

3.1. Process parameter selection based on SLD

Following the original idea of Tobias and Fishwick [313], chatter problems can be solved by constructing of SLD and changing the process parameters depth of cut (a) and spindle speed (N) accordingly.

The SLD is built using a stability model fed with four different inputs: the cutting coefficients or specific forces describing the cutting force, the dynamic parameters of the system, the process parameters and the tool geometry (Fig. 5).

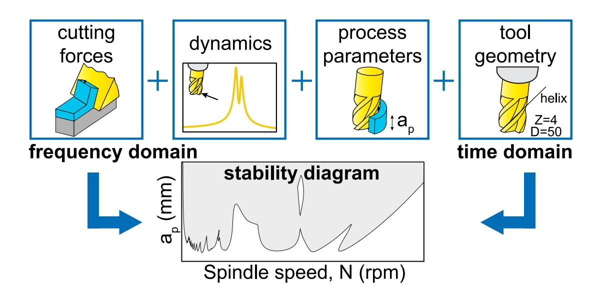


Fig. 5. Procedure to obtain stability lobe diagrams (SLD).

The cutting force is characterized by means of mathematical models fed by *cutting coefficients or specific cutting forces*. Cutting coefficients represent the material's yield strength, friction between the tool and work material, and tool geometry. The cutting force is expressed as a product of cutting force coefficient and chip area, hence it acts as a gain in the closed loop dynamics of the regenerative system. The cutting coefficients are either mechanistically obtained by correlating the measured forces against the chip area by conducting cutting tests with the tool, or by transforming the shear stress, shear angle, and friction coefficient obtained in orthogonal cutting tests to oblique tool geometry [12,30,31,65,103,106].

The *dynamic parameters* of the system are measured experimentally by means of impact modal tests or any of the available experimental techniques [59,109]. The objective of this measurement is to capture the relative vibrations at the tool-workpiece contact zone which are caused by the critical modes of the structure that are limiting the productivity. Depending on the characteristics of the cutting process, the dynamic parameters can vary and, therefore, the identification of several Frequency Response Functions (FRF) might be required at the tool-workpiece contact zone. In large machines, the receptance of the structure should be measured in different positions of the machine [135]. In case of thin wall machining, the dynamic properties can also change as the material is removed. This effect increases the complexity of the dynamic characterization [70].

Process parameters like cutting tool/part engagement are needed for the stability model [111]. The engagement can constantly change in complex operations [112,170], therefore, the stability calculations require a close connection with the CAM system to create a virtual machining environment where the different engagement for different tools, operations and tool paths can be transferred to a stability model [13,18,315] (Fig. 6). In some cases, other process parameters like cutting direction and feed rate should also be defined for stability predictions. This is especially important for serrated cutting edges [98] or when nonlinear force models are considered [106,157,171].

The stability model requires the *geometry of the tool* that affects the cutting force coefficient, the kinematics of the machining operation and the directional coefficients of the dynamic force. The metal cutting processes like turning, milling and drilling are performed with a wide range of tool geometries. Several attempts have been made to propose a unified methodology [19,107,154] based on the discretization of the tool in infinitesimal disks [30,93] and the application of the orthogonal to oblique transformation at each disk [31,64]. These methods make it possible to construct SLD in case of complex tools like the serrated tools [98,200].

Mathematically, the cutting dynamics including the regenerative effect is governed by a delayed differential equation (DDE) with constant parameters (turning) or time periodic parameters (milling). The stability analysis in milling processes becomes more complicated because of the discontinuous nature of the process. Cutting forces are originated by the simultaneous cutting of several flutes with varying magnitude and direction.

Consequently, different models have been developed in the literature to predict chatter by means of SLD (Fig. 7). Altintas and Weck [25] reviewed these *stability models* including turning [83,110,275], milling [14,141], drilling [40,253] and grinding

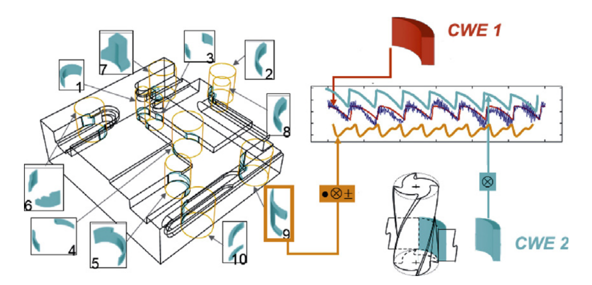


Fig. 6. Tool engagement calculation from CAM model [199].

operations [138,219]. In the last years, different new methods have been proposed for stability prediction. According to their historical importance, the stability models can be classified in four groups: semi-analytical frequency domain based methods [14,312], numerical multi-frequency methods [64,205], initial value time domain methods [281,307] and boundary condition time domain based methods [141,145,319].

The first attempts in dynamic modelling of machining processes focused on obtaining *analytical solutions in frequency domain*. These first single degree of freedom models related the limit depth of cut to the real part of the FRF defining some projection coefficients named directional factors [163,225,312]. These methods were inaccurate, especially for milling operations with multiple modes of similar frequencies. Altintas and Budak [14] proposed a new analytical method, the single frequency method or zero order approximation (ZOA), which is able to solve this inaccuracy in milling processes [107,149]. This method offers a fast estimation of the stability and can be fed directly with FRFs. ZOA is precise enough for most processes, but when highly interrupted cutting processes are considered, inaccuracies due to the presence of an additional stability lobe family related to the double period chatter [87,88,190] and mode interactions [23,218] were found. Nowadays it is possible to have a semi-analytical solution considering the nonlinear feed effect [171] or including an exact prediction o f the double period chatter [217].

Kondo et al. [166] and Tlusty and Ismail [307] proposed to perform *initial value time domain simulations* to study metal cutting processes including milling. These simulations are capable of accounting for nonlinear effects of the process, such as the loss of contact between the cutting edge and workpiece due to high vibration level [22,307], the real kinematics of the process, the change of the cutting force coefficients as the chip thickness varies and the process damping. Initial value time domain simulation

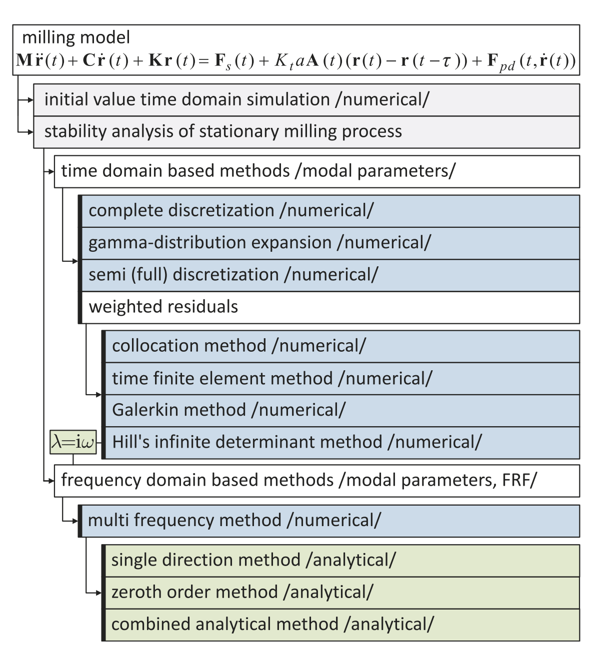


Fig. 7. Classification of stability models.

offers detailed information of the milling process but the stability chart calculation is not straightforward [281] and requires modal parameter extraction from FRF. These simulations do not directly analyse the stability, and therefore they are time consuming. However, initial value time domain simulations are the key to introduce physics based simulations in process planning, and the latest advances in the information and communication technologies are helping to speed up the process for industrial applications [18,44].

In 1993, Minis and Yanushevsky [205] presented a general numerical method for chatter analysis in the frequency domain that was able to handle experimental FRF. It was after this work when the frequency domain solutions came back to the spotlight. These researchers improved the method proposed by Sridhar et al. [284] applying Floquet's theory and Fourier analysis to milling process. Although this *multi-frequency model* relies on the time consuming numerical evaluation of the stability limits, it provides a deeper insight into the milling stability. Double period lobes can also be described in the frequency domain if more harmonics are considered [201,350]. This stability model provides the exact solution but presents limitations to handle complex tool geometries [350,351]. Recently, Bachrathy and Stepan [34] have proposed an extension to handle the dependency of stability on the axial immersion.

The recognition of double period chatter [87] was the trigger for the application of *time domain based methods*. The SLD is completed by scanning different depths of cut and spindle speeds. Different time domain based methods have been applied to solve the DDE associated with milling operations including Gamma-distribution based expansion [141], the semi-discretization [116,142,144], full discretization [96,139], collocation method [73,319], time domain finite elements [39,156] and Galerkin method [255,328]. In general terms, this set of methods in combination with the discretization of the tool and the orthogonal to oblique transformation offer direct stability predictions for complex tool geometries [19,98]. Intricate stability diagrams can also be properly captured including double period chatter and mode interactions [214,266]. These methods require the discretization of cutting and numerical parameters, and therefore the computation time depends on the initial parameter selection.

The stability can be predicted by means of any of the previously mentioned stability models. For instance, if a complex example is considered (Table 1), the multi-frequency and time domain based methods (semi-discretization) are predicting the same result (Fig. 8). The ZOA has problems to predict double period chatter and mode coupling.

This complex example is used to describe the effect of the different chatter suppression techniques in this keynote paper (Fig. 8). The first order semi-discretization method is applied to simulate the effect of the different chatter suppression methods. This method has proven its efficiency to predict the stability of turning [143], milling [145] and grinding [27] processes with process damping [33], complex tool geometries [98,99,140] and continuous spindle speed variation (CSSV) [27,143,347].

3.2. Tool/toolholder selection based on SLD

The optimal cutting conditions are constrained by different factors including the machinability of the material and the power/speed limits of the spindle. The main idea of this approach is to suppress chatter selecting a new tool based on SLD that assures the best spindle speed while considering machinability of the work material and/or spindle power limit at the most stable zone. Depending on the dynamic characteristics of the chatter, this technique can be applied in two different ways:

- *Tool/toolholder selection in high speed machining (HSM) based on receptance coupling substructure analysis (RCSA)*

  The proper selection of the tool overhang and diameter can suppress chatter [311]. This method has been principally studied for aluminium rough milling due to the importance of tool and

Table 1
Simulation parameters.

Tool				
Diameter (D)	Number of flutes (Z)	Helix angle (η)	Lead angle (κ)	
50 mm	4	0°	90°	
Cutting conditions and coefficients				
K_t [N/mm^2]	K_r [N/mm^2]	Radial engagement		Feed direction (**f**)
2000	600	25%	Down milling	(1,0,0)
Dynamic parameters				
Mode	f_0 [Hz]	ζ [%]	k [N/μm]	Orientation (**v**)
1	45	4	30	(1,0,0)
2	60	4	30	(0,1,0)

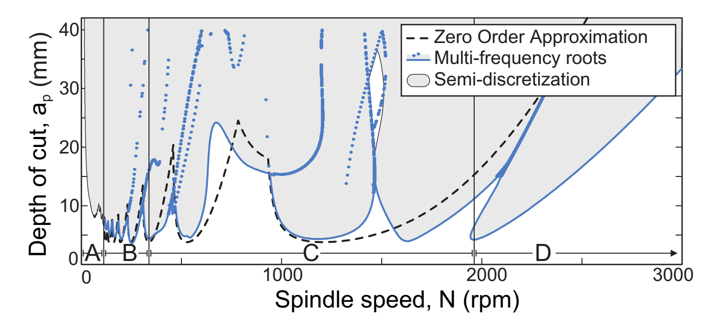


Fig. 8. Comparison between different stability models [218].

toolholder modes in this machining process. By means of the variation of the geometric characteristics of the tool, an increase in the stability of the process can be achieved. In the case of light alloys with good machinability, the tool is selected to have the highest stability point in at the maximum nominal speed [311].

The efficient application of this method requires the combination of the SLD with receptance coupling substructure analysis (RCSA) [188,262,263]. RCSA method allows coupling the theoretical and experimental FRF of individual components to simulate the general response of the assembly (Fig. 9).

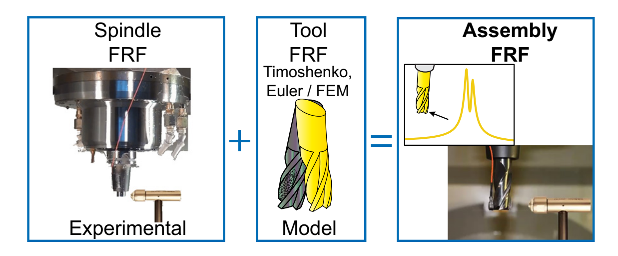


Fig. 9. Spindle/toolholder and tool structures.

In the simplest case [263], two components were joined considering just translational degrees of freedom, while the interface between them was considered flexible. The method was improved introducing the rotational degree of freedom related to bending and torsion with its joint flexibility values [261,262,339]. In recent years, several authors have proposed improvements to this method and have studied the influence of contact parameters [108,236,264].

The major advantage of RCSA is that it can save considerable time simulating the FRF at the tool tip for different tools, combining the theoretical response of different tools with the experimental results of the toolholder/machine assembly.

There are many ways to obtain the theoretical FRFs of the tools. Schmitz [262,263] used analytical Euler–Bernoulli beam theory which is simple and provides reasonable engineering approximations for many problems. However, this model tends to slightly overestimate the natural frequencies, especially for non-slender beams [126]. Timoshenko beam model overcomes these inaccuracies adding the effects of shear deflection to the Euler–Bernoulli's theory. This beam theory was introduced in an analytical model for prediction of tool point FRF [108,233]. Theoretical responses can be obtained by FEM as well [220,236].

The combination of the SLD with RCSA permits the selection of the proper tool to solve a chatter problem. If the experimental point FRFs at the spindle or toolholder level are collected, the effect of different tools can be predicted to optimize the process while avoiding self-excited vibrations.

- *Tool selection for structural chatter cases*

Heavy duty face milling of steel and cast iron is mainly limited by the machinability, spindle power and chatter. Comparing with the High Speed Machining (HSM) of aluminium alloys, the window of suitable spindle speeds for each tool is narrower due to machinability reasons. The objective is to select a tool for which the optimal cutting speed (from machinability point of view) coincides with the most stable zone presented by SLD.

First of all, the machine dynamics must be characterized. When the stability of the cut depends only on the dynamics of the machine structure, the mass and the stiffness of the different possible tools barely affect the dynamic parameters of the critical modes. Therefore, it is justified to consider that the FRFs do not change with different tools, and the RCSA is not needed. FRFs are basically machine position dependent [216], and consequently, the effect of the feed direction should be considered [163,334].

When the machine tool user plans a face milling process, a large number of tools with different diameters and numbers of flutes are available for the same tool insert geometry. All those tools share the same dynamic parameters, cutting coefficients and cutting speed range. Therefore, a common stability diagram can be defined considering the same relative radial immersion and immersion angles [135]. This chart shows the optimum diameter per flute ratio, and the tool providing the highest MRR for the intended process can be selected [135].

Finally, it is important to note that the heavy duty roughing operations can also be carried out using tools and inserts designed for low depths of cut and high feed rates. With this alternative, MRR can be improved, reducing the risk to have chatter (Fig. 10). The axial direction of the tool should be stiff because the inserts introduces large forces axially and the chip thickness has a strong component in this direction.

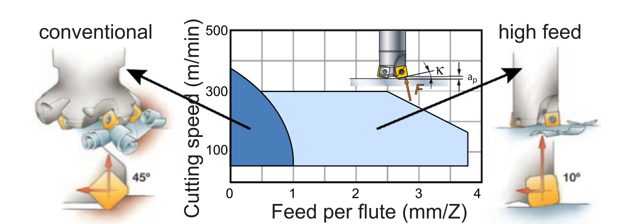


Fig. 10. High feed inserts for chatter suppression.

3.3. Applications

SLD based process planning is useful in cases where chatter arises in low order lobes or high speed range (zones C & D). In these areas, the modification of the spindle speed can increase the limit depth of cut considerably. In industrial applications, milling is the most adequate operation for the application of SLD, because the tooth passing frequency is relatively high and chatter occurs very often in low order lobes.

The main application of SLD in chatter suppression is the HSM of light alloys with high machinability. Several successful milling applications have been reported in the literature related to the HSM of monolithic parts for aircrafts [25, 176]. In case of steel face milling or drilling operations with structural chatter, the self-excited vibrations grow in low order lobes, and the SLD can also be applied to solve the problem. The presence of complex dynamics with multiple dominant modes can limit the improvements [345].

In most turning, drilling and grinding cases, the ratio between chatter frequency and tooth passing frequency is high. In case of high order lobes (zone A & B) at low speeds, the efficiency of this technique is reduced and other techniques are more appropriate for chatter suppression. In the zone ruled by process damping (zone A), the modelling of the stability is extremely difficult, the benefit of the SLD is limited and the obtained information is not robust due to dependency on tool wear. Consequently, the industrial application of SLD for these zones is still limited.

Chatter suppression by SLD based variation of the cutting conditions is time consuming for process engineers. There are commercial software systems (CUTPRO®, MetalMAX®, Shop-PRO®) which are used to compute SLD and simulate the process to assist NC programmers in industry. The developed stability models still have little impact in production plants. This is due to the fact that the models have not been integrated in the most commonly used platforms by process engineers, such as CAM, verification and optimization tools in 3D environments.

Currently, there are several international research groups trying to address this issue [18]. The instantaneous chip thickness computation along the tool is the key point in this investigation. Current CAM and verification software are not capable of computing the chip load according to tool and workpiece geometry. This information can be used to study the stability of the process in each instant. Several products have been launched to the market such as MACHpro® Virtual Machining System and N-PRO® which is a toolbox inside Siemens NX® CAM system. In Germany, the Institute for Machine Tools of the University of Stuttgart (IFW-Stuttgart) works on the coupled simulation of the structure and the cutting process. The University of Dortmund, ISF developed a simulation system to calculate the regenerative tool vibrations for varying engagement conditions [292,293]. The Technical University of Darmstadt has also presented a work in the coupled dynamic simulation of structure and cutting process, aimed specifically for a robotic milling application [2].

3.4. Research challenges

In the last ten years the process stability investigation has been focused on the following topics:

- Process damping models

When the tooth passing frequency is much lower than the chatter frequency (Zone A), the stability lobes dramatically rise increasing the limit depth of cut due to process damping [25].

There is no consensus related to the analytical description of the process damping effect and there are different theories to explain the involved mechanisms [17,110,285]. The most extended one explains process damping as the result of the rubbing of the flank of the tool on the machined surface. When there are many oscillations in a single tool revolution, the effective clearance angle γ (Fig. 11) becomes negative in the slope-down motion of the tool [239,306]. This means that the flank face of the tool rubs and indents on the workpiece surface when the material is being cut, and the contact forces which are against the direction of vibration adds extra damping to the cutting process at low spindle speeds (Fig. 11). Since the modelling of this effect is a complex task, it has become one of the major research subjects. The proper modelling of this process damping is of interest for turning and milling of difficult-to-machine materials which are usually machined in this zone.

Montgomery and Altintas [211] used the contact laws in order to model the penetration mechanism of a harder material in a softer wavy one. This model, as well as other related models, has problems for a proper stability prediction at low speeds. Altintas et al. [17] defined a new dynamic force model for orthogonal cutting with three different cutting coefficients: the static coefficient proportional to the chip thickness, and two dynamic coefficients proportional to velocity and acceleration terms. These coefficients were obtained by means of controlled oscillation tests with the aid of a fast tool servo.

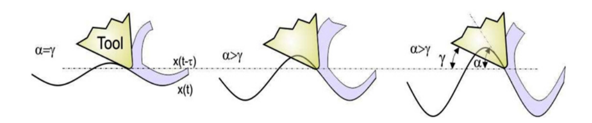


Fig. 11. Indentation between the flank face and the wave form [316].

Nowadays, time domain based methods, like semi-discretization, are regarded as the first method for the modelling of the process damping zone [33]. More recently, Ahmadi and Ismail [7] obtained stability lobes based on a linearized iterative analytical model of the process damping, dependant on the vibration amplitude. Budak and Tunc [69] proposed a new analytical approach to calculate the amount of the process damping acting based on the inverse solution of the stability equation. They introduce an indentation coefficient, which is identified through energy analysis to relate the amount of damping to the indentation volume penetrated underneath the tool flank face. The proposed solution is based on the difference between the theoretical and experimental stability limits. Later on, they extended this approach to milling [317]. They concluded that increased radial depth may not result in significant decrease in stability limits due to the increased process damping [316]. Ahmadi and Altintas [8] recently presented a method of identifying the process damping coefficient from the measured vibrations using operational modal analysis (OMA).

Process damping is, therefore, a very important phenomenon which has not been accurately modelled yet.

- Thin wall machining

In thin wall machining, the dynamic behaviour of the workpiece can become a key factor for the effectiveness of the machining process. This is the case of many parts of the aerospace sector, including structural and engine parts, or pipe threading in oil and gas sector. In these operations, the dynamic properties change as the machining position varies and as the material is removed. The analysis and modelling of the dynamic behaviour of thin walled components is a hot research topic.

One of the first studies of peripheral milling of flexible structures was carried out by Kline et al. [159]. They used FEM to model the plate and beam theory for the end-mill. However, Kline's plate boundary conditions overestimated the stiffness of the part and neglected the effect of the deflected part and the tool in the radial engagement [63]. Budak and Altintas [63] improved this model considering the final surface of the plate. These models accounted only for the static stiffness of the wall.

Altintas et al. [21] simulated the dynamic behaviour of the plate and the tool performing an initial value time domain simulation including the true kinematics. Alternatively, path dependent 3D stability diagrams can be used to design the chatter free thin wall machining [48,303]. Denkena et al. [91] proposed an iterative methodology to simulate errors taking into account the relation between the cutting force and workpiece deformations.

Most of these attempts were carried out in 3 axis operations. However, 5 axis-machining is widely used in manufacturing engine parts like blisk or impellers [68]. Biermann et al. [44] presented a general approach to simulate workpiece vibrations during 5 axis milling of turbine blades (Fig. 12). They combined initial value time domain simulations with dynamic parameters of the thin wall obtained by a FEM model and the surface topology of the final blade was predicted by means of advanced visualization techniques. In one of the recent studies, Budak et al. [70] included the effect of workpiece dynamics on chatter stability in 5-axis milling of turbine blades. They took the continuously varying structural dynamics of the blade type thin wall geometries into account by implementing a structural modification algorithm based on the FE mesh of the final shape and the stock to be removed. They demonstrated that selection of

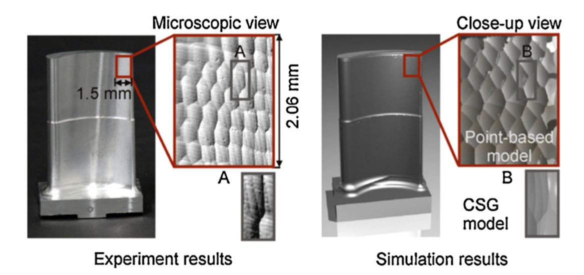


Fig. 12. Measured and simulated surfaces with chatter [44].

spindle speed and cutting depth considering the workpiece dynamics improves both cycle time and process stability.

Similar problems can occur in turning operations. For instance, pipe threading operations for oil and gas applications are performed using special comb-like tools with multiple edges. In these cases, regenerative effects related to the flexibility of the thin wall of the pipe can take place and chatter can arise [286].

Generally speaking, the main tendency in chatter avoidance during thin wall machining is the use FEM models to predict the modal parameters and vibration modes, considering the variations due to material removal. In the meantime, the damping estimation is more difficult due to the effects of the clamping on the different modes of the thin walled workpieces [159,294].

- *Multiple/parallel/multitask operations*

A growing practice in industry is the use of two or more spindle heads machining simultaneously the same or different parts on the same set-up in order to maximize productivity (Fig. 13). However, in some cases, the cutting rate cannot be doubled by using two parallel processes due to chatter [57]. Therefore, for an efficient process with high cutting rates, it is important to determine the stable conditions. In order to predict stability in case of parallel and multitasking processes, it is necessary to understand the dynamic coupling of simultaneously engaging tools via the machine structure or machined part. Parallel turning processes that share the same cutting surface are additionally coupled via the surface waviness [50,66]. This set-up leads to additional difficulty in stability analysis, but gives rise to new chatter avoidance techniques [269].

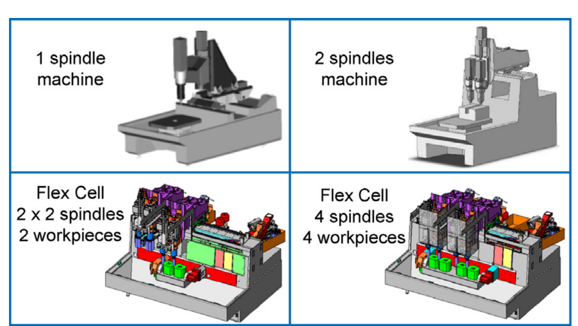


Fig. 13. Machine tool concepts for parallel milling.

Lazoglu et al. [173] studied the parallel turning operations using initial value time domain simulations where each tool was cutting a different surface and dynamic coupling between the tools occurred through the flexible workpiece. They showed that the dynamic coupling decreased the stability limits. Later, Ozdoganlar and Endres [230] developed an analytical solution for dynamically symmetric systems where the cutting tools had the same transfer functions. Ozturk and Budak [66] developed frequency and time domain models for general parallel turning stability analysis. They demonstrated that dynamic interaction between the tools can work in favour of increasing the stability limits compared to turning operations with a single tool. In a recent publication, Özturk et al. [234] differentiated between parallel milling processes with the same cutting surface and those with different cutting surfaces.

Özturk et al. [234] proposed an analytical method for chatter suppression in parallel processes and used it for prediction of stability maps which show stable and unstable zones in terms of depth of cuts pairs of both tools. They took the workpiece flexibility into account in prediction of stability limits which were verified by time domain solutions and experimental results. They also investigated the effects of relative cutting tool dynamics, and demonstrated that the stability limits can be increased significantly by properly tuning their natural frequencies. Parallel milling processes have also been extensively analysed in the last decade. Olgac and Sipahi [223] developed a general procedure to calculate the stability of simultaneous machining including multiple delays. Shamoto et al. [269] worked on the chatter suppression of a double-sided milling process for thin plates. Later, Brecher et al. [57] discussed the time domain simulation of parallel milling processes with two spindles on different workpieces. They focused on the measurement of the so-called dynamic transfer behaviour of the tools, which is defined as the deflection of the second tool due to a force on the first tool. Brecher et al. [56] presented a model for parallel milling considering the dynamic properties of the machine structure using a holistic approach, and mentioned the difficulty of this kind of simulation, since angular relationship between both cutting processes must be considered.

- *Improvement of the SLD accuracy*

An accurate prediction of stability is crucial to suppress chatter based on SLD. During the cutting process, there are several factors that introduce effects that could give rise to stability prediction errors. Exact kinematics of milling process, tool motions outside of the cut, process damping, variable cutting coefficients, unsafe zone [100,270,287] and tool engagement variations due to vibrations have been mentioned as a source of inaccuracies in predicting stability of metal cutting processes (Table 2).

Table 2
Major sources of inaccuracies for SLD.

	Concept	Effect on stability	References
Machine	Gyroscopic effect	Decrease of stability Negligible effect	[208,305,336]
	Thermo-mechanical effects on bearings	Reduction of chatter frequency Decrease of stiffness	[1,76,77,80, 129,181]
	Structure joints	Increase of damping	[133,163,251,268]
	Torsional stiffness	Variations on stability	[20,251,252]
Process	Tool and workpiece contact	Increase of dynamic stiffness	[150]
	Moving of the tool outside the workpiece	Lobe shape remains unchanged	[214,307]
	Process damping effect	Increase of stability	[8,17,33,69,110, 211,239,306,316]
	Variable cutting coefficients	Slight variations on stability	[150]
	Exact kinematics of milling process	Negligible effect	[44,211]

- *New methods to measure dynamic parameters*

Rasper et al. [249] state that machine dynamics are the main source of errors to predict stability. For this reason, current dynamic characterization procedures have been questioned, and new experimental procedures have been proposed (Table 3).

Nowadays, impact hammer testing is the most common method for dynamic parameter identification. Different authors have proposed the use of alternative excitation methods closer to operational conditions. First of all, a shaker [206,249] can be used to study the effect of the force level. The dynamics in rotary conditions can be measured by means of special devices [59,196] or active magnetic bearings. Some authors have also tried to identify the dynamic parameters under cutting conditions using inverse methods, OMA or controlled cutting force variations. They identified deviations compared to traditionally obtained FRFs.

Table 3
Non-conventional methods to measure dynamical parameters.

Excitation	Interest	References
Shaker	Force level control.	[206,249]
Tip testing with rotating spindle	Gyroscopic and the spindle bearing thermo-mechanical effects are considered.	[51,82]
Specific electromagnetic devices		[5,59,196,245,246]
Feed drives	Evolution of OMA.	[177,178,193]
Operational Modal Analysis (OMA)	Real cutting conditions. Direct extraction of modal stiffness is not possible.	[71,72,75] [258,259,346]
Controlled cutting force excitation	All the parameters are controlled. Point FRF is obtained.	[74,136,169,182, 204,227,231]
Inverse methodology	Based on chatter tests. High simplification required. Difficulties for extrapolation.	[3,117,146,158, 168,232,244,295]

4. Special tool geometries for chatter suppression

The cutting process can be stabilized if special tool geometries that perturb the regenerative effect [291] or special edge geometries that increase the process damping are used (Fig. 14).

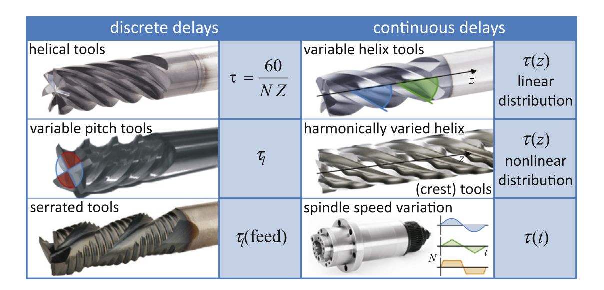


Fig. 14. Delay perturbation strategies against regeneration.

4.1. Special milling tool geometries against regeneration

This section focused on milling processes but the proposed geometries can be extrapolated to other processes with tools of multiple flutes as drilling, broaching, threading and turning with multiple edges [291].

The delay between subsequent cutting edges of any milling process can be directly affected introducing geometric irregularities on the rotating milling tool. Consequently, the introduction of a helical design makes the cutting force smoother, while phases between modulations can be disordered by irregular spacing of edge portions. Discrete variation of regenerative delays can be introduced by simple uneven pitch angles or by serrated profiles in the cutting edge. Finally, different helix angle variations can also produce continuous variations in the weights of the delays (Fig. 15).

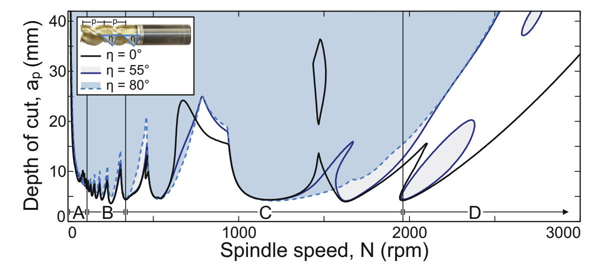


Fig. 15. Effect of the helix.

The cutting force created by these complex geometries can be modelled always by combining the axial discretization of the tool and the application of orthogonal to oblique transformation.

- *Effect of helix*

Conventional helical milling tools have a single constant delay directly related to the tooth passing frequency. The helix introduces an angular delay at the cutting points along the axial axis which soften the force. If a helical shape is introduced in a straight flute, the average of the dynamic force remains constant and the harmonics of the dynamic force are reduced. The average value of the dynamic force is related to the traditional chatter (Hopf bifurcation) and harmonics of the dynamic force create the double period chatter (flip bifurcation) [36,284,350]. Therefore, the helix can suppress chatter in interrupted milling operations dominated by double period chatter [140,238,350].

The efficiency of the applied helix depends on the ratio between the critical depth of cut and the axial distance between two flutes of the tool. In fact, if the depth of cut is equal to the multiple of the axial pitch, the dynamic milling force is constant and double period chatter cannot occur [350] (Fig. 15).

While drilling and turning are intrinsically continuous operations, double period chatter can still appear when special part geometries create an interrupted cutting [296]. In these cases helical edges can also help in the reduction of chatter.

- *Variable pitch tools*

In these special cutters, the flutes are distributed irregularly along the perimeter of the tool. Variable pitch tools, which were first proposed by Hahn [125], disturb the phase between the past and the present vibration. With variable pitch, the constant delay of a conventional tool is modified to create multiple discrete delays depending on the actual edge position and the number of flutes. In general cases, variable pitch tools have a theoretical periodicity related to the spindle rotation frequency instead of the tooth passing frequency of an ideal regular tool.

Slavicek [273] modelled the effect of these tools first and applied orthogonal cutting chatter theory on milling with only two alternating pitch angles. He assumed an infinite cutter radius to avoid the complexity of milling dynamics, and claimed that the maximum depth of cut can be doubled compared to uniform cutters. Similar improvements were predicted using averaged directional factors and experimentally observed by Opitz et al. [226] for the simplest case with two teeth.

More recent studies have shown that variable pitch cutter tools are only useful in high speed zone (zone C) to shift the high stability zones, but they do not significantly increase the absolute stability limit [16,266] (Fig. 16). These tools are more interesting in the intermediate zone (zone B) where this non-regular geometry can increase the stability [291]. While the stability simulations performed with cutting speed dependent process damping do not show any positive effect in the process damping zone (zone A, Fig. 16), experimental improvements in this area with variable pitch tools have been reported [60].

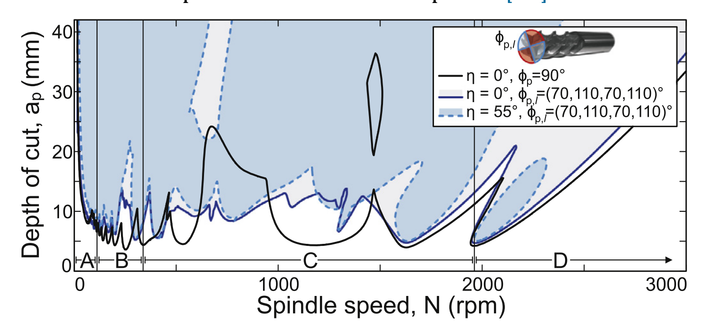


Fig. 16. Effect of variable pitch on SLD.

The selection of the optimal pitch between the flutes is the key to maximize the stability in a certain range of spindle speeds [16,60,291]. Some authors have tried to find simple rules to speed up the design process of a variable pitch cutter. Slavicek [273], for instance, analysed tools with only two alternating delays, and proposed an analytical formula to suppress chatter at certain spindle speed and chatter frequency. These models were based in a simplified model similar to broaching [273,291]. Vanherck [325,326] upgraded Slavicek's technique by considering more than two pitch angles on the cutter, and Varterasian [322] reported experimental results with randomly spaced pitch angles.

Other authors tried to obtain the best design parameters using initial value time domain simulations based on different stability models for variable pitch tools [101,308,309].

The formulation of the ZOA approximation in milling permitted the formulation of faster frequency domain procedures for the optimization of the tool geometry [16]. Based on this approximation, Budak [60–62] introduced an analytical design methodology based on the phase differences between consecutive constant delays. Its effectiveness was confirmed by an industrial application [60].

In the work of Olgac and Sipahi [224], a unique scheme, namely the cluster treatment of characteristic roots paradigm, was used to investigate the effect of two constant delays on the time-averaged dynamics of milling operation with variable pitch cutters. In interrupted milling operations, however, higher harmonics can have significant strength in the regenerative force causing discrepancies in time averaging methods [60]. The application of time domain based stability models can solve these limitations and they can become an important tool for geometry optimization [266,279,348].

- *Serrated tools*

These special tools, first proposed by Strasman [291], have wavy flutes that produce periodic variations in local radii and lead angle. Due to this special profile, the serrated tools cannot be used for finishing operations. The shape of the serration is modelled by a dimensionless function with two parameters: the peak-to-peak amplitude and the wavelength.

Originally this tool was not developed to increase the stability of the milling process. It was only when making experimental comparisons of the chatter performance of multi-tooth cutters that it was observed that serrated tools gave significant chatter improvements [290,291].

The serrated cutting edges can create non-uniform chip geometry along each flute and among the different flutes. The time delay at any point along the cutting edge may differ due to amplitude of the serration wave and feed rate. The process is periodic at spindle rotation intervals. The dynamics of the process are modelled by DDE with multiple delays and time-periodic parameters. Detailed work on geometry was presented by Wang and Yang [331] where the effect of the serration phase shifts between successive cutting edges and stationary cutting force was obtained.

When the chip thickness is smaller than the amplitude of the serration waves, some sections of the flutes do not have contact with the material. As a result, the number of delays increases, and the serrations attenuate the regeneration resulting in increases in the stability limits. If the feed rate increases, the material contact along the serrated flutes increases as well. Thus, the stability limit is reduced and the machining process approaches the performance of regular, smooth end-mills. Complete edge description and detailed cutting force modelling are introduced by Merdol and Altintas [200] for tapered serrated cutters. With proper feed set-up, roughing operations performed by serrated cutters require less torque due to larger local chip thickness values causing less increase in the local specific cutting force. These tools generally have better stability properties [98,200].

Serrated cutters behave as a cutting tool with less equivalent number of teeth depending on the feed. As a rule of thumb, serrated cutters can produce approximately the number of flutes (Z) times higher axial depths of cut than the regular end mills when the feed rate is less than the peak-to-peak amplitude of serration waves on the cutter.

Serrated edge profile optimization method was shown in [164] where the stability properties and force load were also taken into account. Additional constraints, such as serrated edge failure and flank wear caused by large chip loads, need to be considered when selecting the efficient feeds of serrated end mills used for roughing difficult-to-machine alloys. The serration changes the high stability zones according to the feed rate. Experimental results were presented in [289] where extra resonance stability pockets were confirmed (Fig. 17).

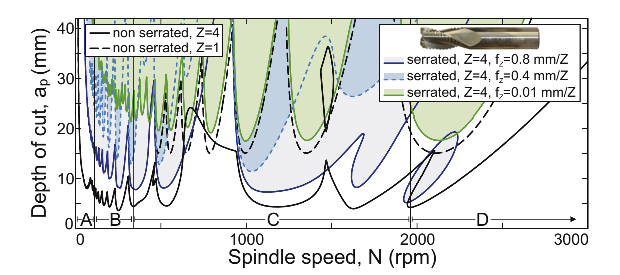


Fig. 17. Effect of the serrated edges on SLD.

The major disadvantage of serrated cutters is the quality of the surface finish that is far from flat because of the waviness of the cutting edges. Consequently, in an attempt to produce improved surface finish, some manufacturers introduced trapezoidal profile cutters [243]. The properties of these cutters are similar to the serrated tools, but the reduction in the regenerative force is far less than for a serrated profile [291].

- *Variable helix tools*

Helical tools with non-constant or alternating helix angles introduce continuous changes in the local pitch angles along the tool axis [290,326]. This special geometry causes continuous variation in the delay between flutes, which then perturbs the regenerative effect (Fig. 18).

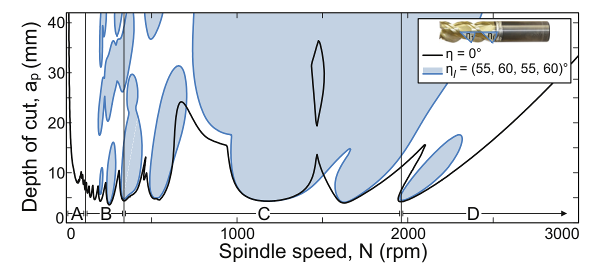


Fig. 18. Effect of variable helix.

The stability prediction of these tools is difficult due to the variable delays. The first predictions were obtained by simplifying the dynamics [318] and using equivalent variable pitch representation of variable helix tools [309].

The axial discretization of the tool is required for more accurate stability predictions. Time domain based methods such as time domain finite elements and semi-discretization were used to build general models of variable pitch and helix tools [99,279]. Optimization processes based on these models were established by a genetic algorithm [342,343].

For variable helix tools, time domain based methods predict important stability variations in the low order lobes (zone A & B) with the fragmentation of the unstable zones into several islands [140]. These results require further experimental confirmation.

Finally, harmonic helix variation has been theoretically investigated using multi-frequency solution [229] and semi-discretization [99]. However, the application of this geometry is residual and the continuity of the chip is not completely assured.

4.2. Special tool positioning in parallel machining operations

In parallel turning similar effect can be created by optimal arrangement of the cutting edges when several tools are operating on the same surface [66] (Fig. 19).

If the relative position of the edges is optimized, the regeneration between consecutive edges can be disturbed. Brecher et al. [50] showed that chatter can be suppressed by tuning the

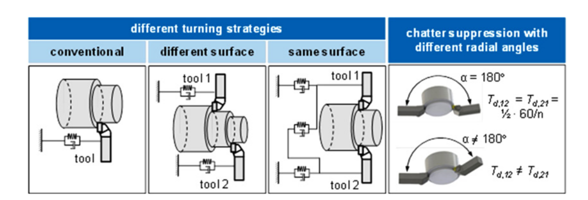


Fig. 19. Chatter suppression for parallel turning [50].

angular positioning of the tools which cut the same surface. Simulated and experimental results confirmed that a dependency of the process stability limit on the radial angle is only evident if the dynamic coupling of both tools via the machine structure is taken into account. Parallel turning machines have this coupling when both turrets are attached to the same column.

4.3. Edge geometries to increase process damping

When the chatter occurs in zone A, three different alternatives can be used to take advantage of the process damping effect.

Firstly, the spindle speed can be reduced to increase the number of vibration waves per revolution and increase process damping. However, this technique causes reduction in productivity and it can be limited by machinability constraints.

Secondly, worn tools can be used to avoid chatter as the process damping increases with flank wear. Chatter often appears when the tool is new, while it disappears later when wear progresses. This effect is stronger when the ratio between the chatter frequency and the tooth passing frequency is higher. However, worn tools increase the static cutting loads and excessive wear may lead to tool breakage and poor surface finish.

Finally, special edge geometries can be used to avoid sharp edges and reproduce the performance of a worn tool. Several authors [316,344] studied the effect of cutting edge geometry on process damping. They found that the increase of cutting edge radius tend to increase process damping. Flank face geometry, clearance angle and drop distance have also strong effects on process damping and chatter stability at low cutting speeds. It was observed that cylindrical flank geometry results in increased process damping compared to planar flank geometry [316].

4.4. Applications

Special tools have been used to suppress chatter and maximize productivity, and they are well established in industry.

Serrated tools used in the right feed range reduce the risk for chatter in roughing operations.

In general terms, variable helix tools can considerably increase the stability in high order stability lobes (zone A & B). Helical tools can remove double period chatter in interrupted cutting operations if the axial pitch is in the range of the depth of cut.

Variable pitch cutter tools are only able to move the high stability zone in low order lobes (zone C), but they do not increase the absolute stability limit [16]. Consequently, they can be used in high speed zone (zone C) when the machinability of a material limits the cutting speed range, and the optimum spindle speed coincides with a low stability zone. In low order lobes, the different pitches of the tool can be tuned to have optimal stability and machinability in the same spindle speed range (Fig. 16) [16]

Chatter suppression is not assured with any pitch distribution, and a proper tuning of the pitch angle for the dominant mode is required.

4.5. Research challenges

Four main research challenges are identified in this chapter to further improve the chatter-resistant multi-tooth cutters [291]:

The optimization of the tool geometry including serrated flutes, variable pitch and helix is supported in time consuming heavy simulations. There are procedures based on energy criterion and the minimization of the regenerative effect, but there is a need for a set of clear design rules to maximize the stability.

Recently, interesting properties have been found for variable helix tools [279]. An experimental verification of stabilization of high order lobes is required for further developments.

The effect of special tool and edge geometries in process damping needs further investigation. Some works [316,344] have tried to clarify this aspect but it would be helpful to find clear analytical rules to enhance process damping.

Finally, the special tool arrangements for parallel cutting processes are a promising chatter suppression technique that requires further investigation to find practical design rules.

5. Increase of stiffness

The stiffness of the machine tool structure, tool and/or part can be increased to suppress chatter efficiently. If chatter produced by one dominant mode is considered, the static stiffness and modal stiffness are similar, and its increase improves process stability. However, the machines have more than one mode with varying directions. The static flexibility is the sum of the flexibilities of all the modes. Therefore, the improvement of static stiffness does not assure the solution to the dynamic problem. In some cases, such as bolted or welded joints, an augmentation of the static stiffness can entail a damping reduction [163].

5.1. Methods for redesign

In this section, methods to redesign the manufacturing system to increase the stiffness for chatter suppression are highlighted. The methods are listed in the order of their complexity since one method includes the test proposed by the previous ones (Fig. 20).

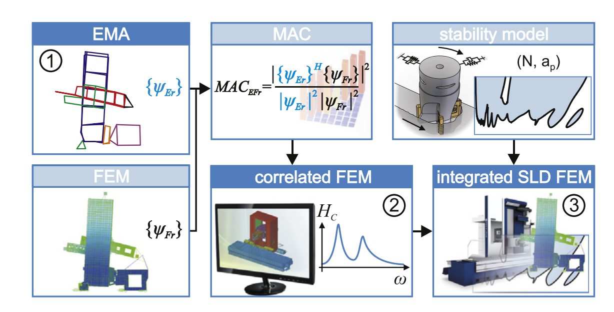


Fig. 20. Theoretical and experimental model correlation.

- <u>*Experimental modal analysis (EMA)*</u>

  In order to avoid chatter, Merrit [202] proposed to perform cutting tests once the machine had been built and then carry out changes to tackle problematic modes of the machine. The most important point is to identify the critical mode that is involved in the self-excitation. This identification is performed by measuring the chatter frequency and comparing it with the different experimental receptance measurements of the tool and the part.

  The chatter frequency can be determined by means of accelerometers or microphone and performing a Fast Fourier Transformation (FFT) to obtain the frequency content of the vibration signals. Chatter usually appears as relevant peak in the frequency domain, which does not coincide with the harmonics of the tooth passing frequency [244]. Tests in different cutting conditions are helpful to clarify the presence of chatter.

  The experimental receptances at the tool and workpiece sides are mainly measured with impact modal tests, preferably in three Cartesian directions, to identify the weak modes.

  The critical mode can be identified with the comparison of the chatter frequency and the different FRFs. In most cases, the chatter frequency is close to one of the most flexible modes of the system. However, in some cases the stability can be limited by more than one mode and chatter frequency can be located

between two modes or can have important modulations close to two different modes [218].

Once the critical mode or modes are identified, Experimental Modal Analysis (EMA) is performed to obtain the mode shape of the corresponding critical mode [163,332]. Finally, the modal stiffness of the critical mode can be increased considering this mode shape. The origin of the motion where the strain energy is high is reinforced to increase stiffness and stability efficiently.

- _Correlated finite element models_

  The previous method can be enriched using theoretical machine models built from numerical procedures like FEM. If theoretical and experimental models are available, their correlation can be checked by means of a Modal Assurance Criterion (MAC) and the natural frequency deviations. The MAC matrix is built comparing the theoretical and experimental mode shapes [109].

  If the correlation is high and the FEM model represents properly the experimental critical mode with a deviation lower than 15% in frequency and a MAC value higher than 0.8 for the correlated mode and less than 0.3 for the rest of the modes, the experimental damping is assigned to the numerical model [109,115]. Later on, the FEM model is used to introduce design variations with the purpose of increasing the stiffness of the critical mode. In this situation, the FEM model can highlight the zones with higher strain energy and the optimization can be performed to increase the modal stiffness of this mode.

- _Integration of stability lobe diagrams (SLD) in finite element models (FEM)_

  The introduction of the SLD in the modal stiffness optimization loop is the next step ahead. A successful correlation procedure permits to assign experimental damping ratios to modes obtained by numerical models. Therefore, the theoretical receptances can be obtained from different design variations, and the effect on stability can be predicted directly based on the stability models mentioned in Section 3.

  With this procedure, the effect of a design variation and the stiffness improvement in stability are linked directly. This design tool permits to close the design loop when the desired chatter-free productivity is obtained by means of increased modal stiffness.

5.2. Fixtures and stiffeners for thin wall machining

Workpiece flexibility can lead to chatter like in case of thin walled aeronautical parts, pipes for oil & gas industry or large welded frames in heavy duty machining. This problem is essentially difficult because the workpiece dynamics vary with both the material removal and the cutter location. Moreover, for flexible workpieces, the static deflection and the forced vibration created by the stationary part can be a major problem. Different chatter suppression techniques can be applied to thin wall machining including process planning by means of the combination of SLD and FEM models to predict the evolution of the dynamics of the thin wall [44,46,48,70,78,122,152,189], application of special tool geometries [60], parallel milling strategies [269], passive [47,167,338] and active [355] damping techniques. In this section the strategies targeting the increase of stiffness of the wall are reviewed.

The compliance of thin walls can be reduced by introducing stiffeners to support the flexible parts at the weakest points. The stiffeners are usually part of special fixtures designed to hold the part. Similarly, special structures are designed to increase the stiffness of pipes to suppress chatter in threading operations. These supports are dynamically effective, but they can create dimensional problems if an excessive force is applied to the thin wall. The design of this specific fixture was studied by Zeng et al. [353] by optimizing the location, the applied forces and the number of support elements. Aoyama and Kakinuma [29] proposed to use a multi-pin supporting system. Some manufacturers have introduced rubber diaphragms and air/fluid balloons to increase the dynamic stiffness of the thin walls avoiding important static deformations [121].

Several research projects try to overcome these problems by using active stiffeners with force and displacement control.

The thin walls can be stiffened by introducing some temporary or sacrificial structures joined to the weak nodes of the wall. Additive technologies can be used for this purpose. The most used material for temporary stiffening of a thin wall is bee wax. In some cases, the different chambers around the thin wall are filled by wax or different thermoplastic materials using their low melting temperature. However, this solution is difficult to automatize. Subtractive technology can also be used to create sacrificial structures to increase the stiffness of a thin walled part. Smith et al. [283] used sacrificial structure to stiffen the part during roughing and semi-finishing (Fig. 21). Those structures created during the primary process or by the machining process itself are removed during the finishing operations.

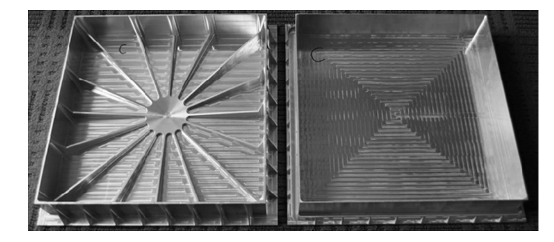
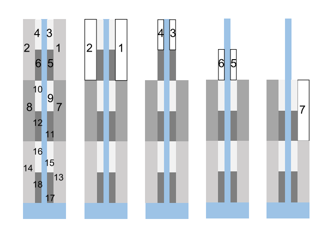


Fig. 21. Sacrificial structures for thin walls and thin floor [283].

The stiffness of the thin wall is also related to the material removal sequence defined in the process planning. Thus, the machining strategy can also have a tremendous effect on the quality of flexible workpieces. Several strategies have been proposed to maximize the stiffness of thin wall during the machining process [9]. HSM strategies with small radial depth of cut, large axial depth of cut and high cutting speed help to reduce the thin wall deflections. For the machining of thin ribs, typically found in aeronautical structures, expert knowledge is used to define the maximum axial depth of cut depending on the remaining thickness of the thin wall and to choose the order in which the material should be removed. The four-to-one rule (4:1) for aluminium and the eight-to-one rule (8:1) for titanium ensure that the stiffness of the thin wall is high enough to withstand the cutting forces and also to avoid chatter [192] (Fig. 22). This method has been developed by Boeing Manufacturing R&D group who called it as the waterfall machining technique [352].



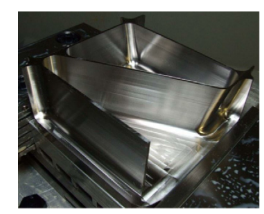


Fig. 22. Waterfall strategy for thin wall machining [192].

Smart tool path allows maintaining the workpiece rigidity. The machining process of thin walled airframe components has been optimized by means of the material removal sequence [165] and the tool orientation [172] to minimize the workpiece deflection.

5.3. High performance materials to increase the stiffness

Chatter problems can be solved in some cases by changing the material of some elements of the mechanical system. Once the critical mode is identified, the introduction of materials with high Young's modulus in zones where the strain energy is maximum can increase the stiffness of the critical mode. These changes can be combined with the introduction of light materials in points with high critical modal displacement to increase natural frequencies. However, this kind of pure increase of natural frequency leads to the variation of the location of the stability pockets, and does not lead to a direct improvement in stability.

Steel and cast iron are dominant materials in machine tool structures, spindles and fixtures [207]. Only costly materials like carbide and special metal alloys and composites can offer substantially larger Young's modulus [191]. Therefore, the material variation is an alternative only in local and high frequency chatter cases of small elements like tool and boring bars. For instance, Lee and Suh [175] reported an important stability improvement by means of a graphite epoxy composite boring bar [298]. In these cases, the use of carbide may be economical to ensure stability. Rivin and Kang [252] proposed the use of multi-material modular boring bars for stability enhancement. The base of the boring bar was formed by a carbide module to support high strain energies and the tip was made of reinforced carbon fibre.

5.4. Applications

The enhancement of the stiffness of the critical vibration mode is always a reliable solution to suppress chatter problems. However, two aspects limit the suitability of this solution: the cost of the implementation and the space restrictions.

The stiffness enhancement of the critical modes is an appropriate solution when the process variations are not feasible. This situation occurs when the chatter is produced in universal machine tools where many different machining operations can be programed and some of these operations are highly limited by structural chatter. In this situation, the user can select any problematic cutting condition and the increase of the stiffness of the critical modes is one of the best solutions. The same solution can be adopted when the cutting conditions and tools are frozen and there is no option for process variations.

In addition, lower the natural frequency is, the more global the critical mode is and the higher the cost for stiffness enhancement is. For instance, the structural chatter problem usually induces high cost variations for chatter suppression once the machine tool is built. On the other hand, in high frequency chatter problems, strengthening the critical mode can be applied in the case of chatter related to tool, boring bars and thin walls.

Nowadays, there is no design framework combining FEM calculations and stability predictions for chatter suppression and design optimization. Only the SPINDLE-PRO© application permits to predict the stability of different design variations of the spindle head, toolholder and tool.

In the case of chatter produced by the flexibility of the part or thin walls, another important point, is the number of parts to be produced. If large batches are produced, the design of special fixtures with special stiffeners can be a viable solution to avoid chatter problems. Otherwise, the design and fabrication of a special fixture can be too costly.

5.5. Research challenges

A holistic optimization tool to suppress chatter and improve the process stability of machine tools in the design phase is required for the future.

The stability simulation requires precise data related to the damping of the system that is originated in the joints of the manufacturing system. The modelling of the dynamic stiffness of different joints is one of the most difficult challenges in the way to predict the dynamic behaviour in the design phase.

Many researchers focused on the development of passive and active fixtures for thin wall machining [355]. The development of reliable and cost effective active fixtures that are able to strengthen the thin walls with controlled deformation is an interesting alternative research direction.

6. Passive damping techniques

The objective of these solutions is to increase the damping of the critical mode by passive solutions without any external power supply for vibration energy dissipation. Damping is a concept to describe a group of different physical mechanisms that dissipate mechanical energy. Various passive damping strategies can be used to suppress chatter. In this section only two possibilities are discussed: the passive auxiliary systems (Fig. 23), and application of special materials and coatings.

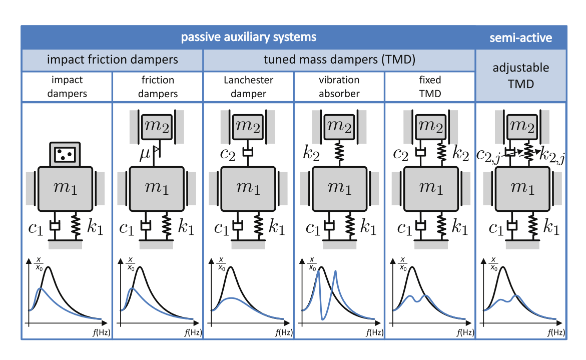


Fig. 23. Passive auxiliary systems [49].

6.1. Tuned mass dampers (TMD)

The original idea was introduced by Hahn [124] when he proposed the use of a Lanchester passive damper with the objective of suppressing chatter.

- _Basic concept_

  A TMD is an inertial mass added to the system to damp via a linear spring of stiffness k_2 and damping c_2 (Fig. 23). The values of these parameters are tuned to damp the critical mode of the original system that may produce chatter. By matching the natural frequency of the highly damped TMD with the critical frequency of the system both modes can be coupled to increase the damping. Consequently, the original mode is split into two modes with high dynamic stiffness (Fig. 23). The TMDs have to be tuned accurately to the targeted frequency, and their positive effect is limited to a certain frequency range [89,323]. The great advantages of the TMDs are their simplicity and reliability.

  The TMD should be located in places where the critical mode has large modal displacements. This way, the equivalent mass of the original critical mode is lower [109], and the required mass of the damper can be reduced for the same mass ratio μ (Table 4). In general terms, the critical modes have large displacement close to the cutting point where the available space is small. The application of high density material like lead or carbide can help in this search for compromise between weight and space.

Table 4
Dimensionless parameter of TMD.

Mass ratio	Damping ratio	Frequency ratio	Natural frequency
$\mu := \frac{m_2}{m_1}$	$\chi := \frac{c_2}{2m_2\omega_1}$	$f := \frac{\omega_2}{\omega_1}$	$\omega_2 = \sqrt{k_2/m_2}$

  The viscous damping can be provided by means of viscous fluid dampers [324], squeeze film damping or eddy currents [6,267]. Alternatively, the high structural or viscoelastic damping of some materials can be used to build a TMD [349]. In this case, the damping is considered to be proportional to displacement instead of vibration velocity by defining a loss factor and a complex stiffness. Finally, piezoelectric materials passively shunted with resistors can also provide structural damping [195]. These materials produce a voltage under strain and this electromechanical coupling can be used in passive vibration control to dissipate energy.

  Two special TMDs can be considered: the Lanchester damper when the mass is connected via a dashpot only ($k_2 = 0$) and the vibration absorber linked to the original system only by means of a spring ($c_2 = 0$). In the last case, the vibration absorber divides the critical mode into two, but it does not directly increase the damping of the critical mode. Therefore, it is not useful for

chatter suppression. On the other hand, different variations of the Lanchester damper have demonstrated their efficiency against chatter [124]. Practically, the Lanchester damper can be built submerging a mass inside a cavity which is filled with a viscous fluid [163]. When the vibration of the mass occurs, the fluid is forced to flow through a small channel between the casing and the body. This way, the damping value depends on the gap between the mass and the cavity and the viscosity of the fluid. Some authors have used eddy currents to build a similar concept [338]. Theoretically, the efficiency of a pure Lanchester damper is lower than a complete TMD [312], but it can affect other modes and have a wider broadband [338].

In recent years, a number of more complex passive damping systems, such as dampers with multiple degrees of freedom [358] and multiple TMDs [337], have been developed to damp a single mode. It has been demonstrated that multiple TMDs are more effective than a single TMD solution having the same mass ratio.

• *Optimal tuning*

In the tuning procedure, the values of the mass m_2, stiffness k_2 and damping c_2 are selected to damp the critical mode. The first step is always to obtain the equivalent modal mass m_1, and the natural frequency ω_1 of the original critical mode at the selected location for the damper. In the second step, the mass ratio is determined considering the available space and other physical constrains. In general terms, the higher the mass ratio μ is, the more robust the TMD is against dynamical variations. The TMD can be efficient with mass ratios around 4% [337], but for industrial applications higher mass ratios are recommended. For instance, in case of damped boring bars, high mass ratios are used to have a robust solution (up to 20%).

For general vibration problems, the classical tuning strategy was developed by Den Hartog [89]. However, this classical tuning methodology is not an optimal solution for chatter. In the simplest case, the chatter free critical depth of cut of a machine is inversely proportional to the real part of the FRF at the tool-workpiece interface [310]. Instead of targeting a reduction of magnitude, Polacek and Outrata proposed the reduction of the real part of FRF by designing a proper passive damper [163,240]. Following this idea, Sims obtained an analytical expression for the optimal tuning for chatter [276] (Table 5).

Table 5
Optimal tuning of TMD.

Target	Frequency ratio	Damping ratio
Receptance magnitude [58,89]	$f_{DH} = \frac{1}{1+\mu}$	
Receptance real part [276]	$f_{SIMS,p} = \sqrt{\frac{\mu + 2 + \sqrt{2\mu + \mu^2}}{2(1+\mu)^2}}$ $\omega_c > \omega_1$	$\chi \approx \sqrt{\frac{3\mu}{8(1+\mu)^3}}$
	$f_{SIMS,n} = \sqrt{\frac{\mu + 2 - \sqrt{2\mu + \mu^2}}{2(1+\mu)^2}}$ $\omega_c < \omega_1$	

In actual machining operations, the stability is often affected by more than one mode with certain damping levels. Therefore, the analytical tuning formulas do not provide the optimal solution, and numerical optimization formulas are used to obtain the optimal solution. In the meantime, the use of the Den Hartog's or Sims' formulas as an initial guess assures a fast convergence of numerical algorithms. For instance, Yang et al. [337] applied the minimax algorithm to optimize multiple TMDs.

• *Semi-active tuned mass dampers and self-tuning algorithms*

The efficiency of TMD is limited if the dynamics of the original system vary due to changes in the position of the tool in the workspace or due to the material removal of flexible parts. In this situation, Slavicek and Bollinger [274] proposed the application of semi-active dampers for which the natural frequency and the damping of the system can be varied.

Semi-active dampers require some energy to be tuned but after that the system behaves like a passive damper. The energy consumption is low compared with pure active devices and a small battery is sufficient for that purpose [215].

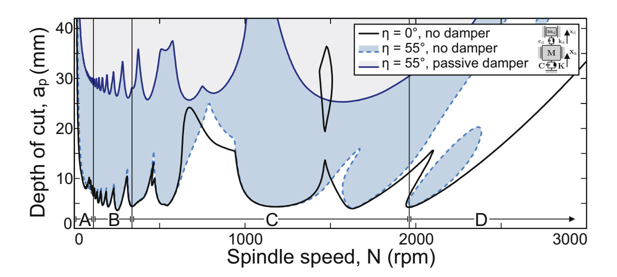


Fig. 24. Effect of TMD in the stability.

Although the proposed strategies are not much complex mathematically, the achievement of an accurate tuning is challenging. The control of the tuning requires a mechanism that can change at least the natural frequency ω_2 of the TMD. The values of the spring stiffness k_2 can be changed by means of squeezing an elastomeric O-ring, creating a relation between the preload of the ring and the tuning frequency [274]. However, damping and stiffness are fully coupled in the ring and cannot be set independently. Seto [267] solved this problem by changing the contact point of the moving mass along cantilever beam. The viscous damping was provided in parallel by means of fluids or eddy current effect. Recently, a self-tuned TMD was presented [215], where the stiffness is tuned by means of a rotary spring and damping is provided magnetically.

There have been several attempts to control the amount of damping actively by means of electrorheological (ER) [28,330] and magnetorheological (MR) fluids [94,203]. For instance, Díaz-Tena et al. [94] were able to damp thin floor parts using a MR fluid. The TMDs based on shunted piezoelectric systems can be used to create semi-active dampers as well [195].

A self-tuneable TMD also requires an automatic tuning procedure. An iterative algorithm based on the chatter frequency determination has been proposed by Munoa et al. [6,215].

6.2. Friction/Impact dampers

Friction dampers use the frictional force that results from the relative movement between the vibrating element and the added mass or the different parts of the damper.

For example, damping was increased by inserting a plate slightly larger than a rectangular hole inside a boring bar [194]. Friction between hole and plate during bending vibration results in energy dissipation, leading to the improvement of the damping and stability of the tool. Edhi and Hoshi [102] proposed a frictional damper to eliminate the high frequency chatter in boring. The proposed damper had a simple structure consisting of an additional mass attached to the structure via a piece of permanent magnet. Ziegert et al. [356] explored the possibility of deploying a mechanical damper directly inside a rotating cutting tool. The damper consists of a multi-fingered cylindrical insert placed inside a matching axial hole along the centreline of the milling cutter (Fig. 25(b)). When the tool is rotating during the milling process, the centrifugal force creates a pressure between fingers and tool body leading to energy dissipation via friction work during bending vibrations. This damper improves the efficiency of the tool up to 53% in the best case.

On the other hand, impact/shock damper dissipate energy when two bodies impact, and it is not negligible even in steel bodies. This principle is the base of the impact dampers. In machine tools, it was first applied for chatter suppression in turning by Ryzhkov [254], and the application in boring bar is also well known [105,304]. It consists of a cylindrical mass located directly in a cylindrical hole at the end of the bar. The design is very simple, but in order to attain maximum efficiency it is necessary to find the optimum value of gap between the cylinder and the hole. The length of the cylindrical mass is usually taken as two times the

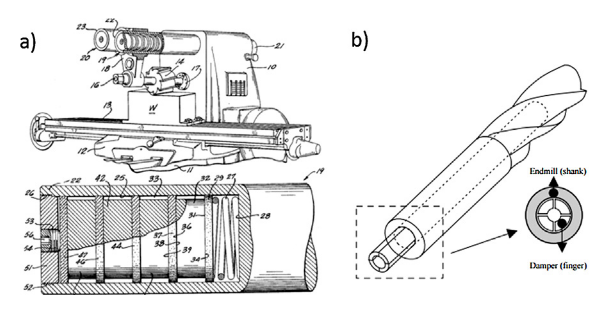


Fig. 25. Friction/impact dampers: [228,356].

diameter [163]. The space is filled by air only. The same principle can be extended to create granular or particle dampers where a bunch of elements are located in a cavity to add damping through impacts among particles [277].

Friction and impact dampers are often combined to maximize vibration dissipation. For instance, Osbum [228] divided the body of a damper into a set of disks that are compressed by a spring. When the system vibrates, impacts occur, but there is also relative movement between the disks and, consequently, friction provides an additional source of damping (Fig. 25 (a)).

In many cases these systems are set up based on a trial and error procedure. Friction and collision are difficult to simulate, even though some authors have already worked in this field [43,160].

6.3. Introduction of materials with high internal damping

The introduction of special materials, coatings and foams with high internal damping has been proposed by several authors to suppress chatter [167]. These materials should be placed at the locations where the modes have the maximum strain energy. For instance, the utilisation of high damping metallic materials (hidamets) has been proposed [298,321]. Previous cases have shown the positive effect of the enhancement of damping of the critical mode. However, the main source of damping is at the interfaces and joints of the system. This damping is one order higher than the structural damping of metallic parts [163,268].

In case of structural chatter with low frequency modes ($f_c < 200$ Hz), this is especially true, and damping is fully defined by joints. In fact, the selection of the guiding system of the machine is more important for damping than the used materials [163,251]. Rashid and Nicolescu tried to increase the damping by creating highly damped interfaces for workpiece holding systems [134,248], turning and boring bars [86].

The high damping materials are especially suitable in high frequency chatter cases dominated by local modes with low damping, where the damping originated at the interfaces is low. Viscoelastic materials can be sprayed or bonded over the surface of the vibratory structure. The appropriate application is the thin wall machining. Kolluru et al. [167] applied a viscoelastic layer (neoprene) in the thin wall, and noticed that this flexible sheet only damps the high frequency content in the vibration signal due to its dynamic properties leaving the low frequencies undamped. In the dynamic impact testing trials, the peak response of the undamped casing was reduced after applying the neoprene layer. The cutting tests corroborated the dynamic improvement. Finally, some authors have introduced coatings that are able to damp high frequencies on panels, sheet metal structures and disk shaped tools [113,314].

6.4. Applications

Passive techniques are able to enhance the damping of the critical mode and, therefore, the absolute stability limits are increased. In general terms, it is a solution with positive effects in all the zones of the stability lobe, and it can always be a solution for chatter problem (Fig. 24). It is especially useful when the low machinability of the material does not permit any change in process conditions, when the damping of the original system is low or when spindle speed variations do not assure a sufficient stability improvement (zone B).

The use of passive dampers in machine tool structures and work holding systems should not be considered as an emergency solution. It often permits the attainment of a larger increase of stability of the machine tool against chatter than by increasing the stiffness which is in some cases leads to large increase of weight [163]. However, the machine tool industry shows reluctance to introduce TMDs in comparison with other sectors.

Passive dampers are mainly applied in slender boring bars where the geometry of the part does not permit the increase of the stiffness [163,339]. When the slenderness or the overhang to diameter ratio is higher than 5, the introduction of these devices is essential to have a chatter-free surface. There are many products based on passive damping techniques including variations of the TMD for turning and milling applications (Silent tool® from SANDVIK, KENNAMETAL, Steadyline® from SECO, etc.), and damped viscoelastic tool holders (ZVT® from MIRCONA).

6.5. Research challenges

The reluctance of the industry applying TMD is related to its main drawbacks: large space is required at critical locations, the effect is limited when dynamic properties vary, and the tuning procedure is based on unusual techniques in industry. The development of self-tuneable compact and robust TMDs with a reasonable cost can help to open the doors of the factories to these devices, and repeat the success of damped boring bars.

Damping control will improve in the next years due to the application of new fluids that can change their viscosity depending on magnetic or electric field or by the active application of eddy currents. The shunted piezoelectric systems also offer new possibilities for passive and semi-active damping.

Friction/impact dampers are simple solutions which require smaller volume than ordinary TMDs. The lack of design criterion and simulation tools for these dampers is an important drawback for their application in industry.

Thin walls machining is also an interesting niche for passive damping techniques due to the low damping of the critical modes. The development of an industrial and automated passive solution for chatter suppression respecting the surface integrity of the parts is still a challenge.

7. Spindle speed variation techniques

In this section, the strategies based on the variation of spindle speed will be reviewed. All these techniques try to change the tooth passing period discretely or continuously in order to vary the period between the modulations in the chip thickness, which are creating the regenerative instability.

7.1. Discrete spindle speed tuning (DSST)

The automatic stable rotational speed selection or regulation technique is a chatter avoidance strategy developed in the early 90s. However, it has been exploited mainly in the production of monolithic aluminium parts, but its scope is much broader. Smith and Tlusty [280] described the basic theory behind the existing system for chatter elimination in milling, through the automatic adjustment of the spindle speed. The system described there does not require the knowledge of system dynamics and selects iteratively a stable spindle speed.

This discrete spindle speed tuning (DSST) technique is based on two hypotheses that are true for one dominant mode: high stability pockets are related to resonance conditions for one of the harmonics of the tooth passing frequency, and the chatter frequency f_c is close to the natural frequency of the critical mode. DSST proposes a new spindle speed N_{ss} which can be tuned close to the programed speed with the help of the lobe number k (eq. 2), the

number of flutes Z and chatter frequency f_c.

$$N_{SS} = \frac{60 f_c}{Z(k+1)}. \quad (3)$$

The method tries to find the resonance iteratively based on the experimental measurement of the chatter frequency f_c. In each step, the spindle speed N_{ss} is changed to make one of the harmonics of the tooth passing frequency coincident with the measured chatter frequency. When the new spindle speed is set up, chatter may disappear or its frequency changes providing the input for the next iteration. The procedure theoretically finishes when the chatter is eliminated or the chatter frequency is not changing after two consecutive iterations [280]. Once the stable cut is found, the depth of cut can be increased by repeating the iterative process. This process can be traced iteration by iterations in the stability diagram (Fig. 26).

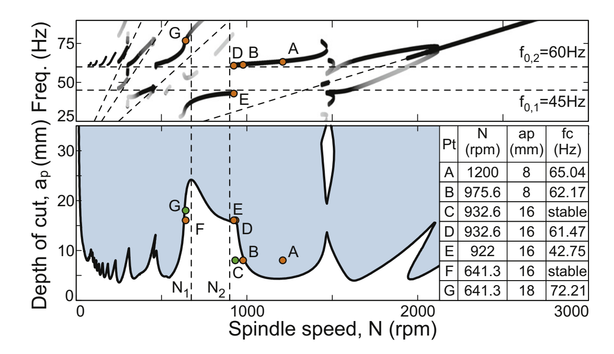


Fig. 26. Iterative procedure for discrete spindle speed tuning.

At first glance, it may seem odd to try to solve a vibratory problem by finding the resonance, but this assures that the modulations that affect the chip thickness are in phase to avoid the regenerative effect. In fact, the sweet spot is close to, but not exactly at, the resonance to avoid high forced vibrations.

This procedure can be applied off-line using a learning procedure for the cutting process where chatter frequency is measured and the new spindle speed is specified. Alternatively, it can be used on-line by varying the spindle speed automatically according to the iterative algorithm [180]. The drawback of the on-line strategy is that the spindle speed is varied automatically and can cross unstable zones during this process [333].

The on-line procedure requires an algorithm for chatter determination. Several methods have been proposed in the literature [244]. The chatter frequency can be measured by means of microphones [90] or accelerometers [41]. In low frequency chatter internal signals of the CNC can be used as well.

It should be noted that if the natural frequency of the main mode is obtained by means of a tap testing or with an ordinary impact, the spindle speed can be directly defined close to the resonance of the main mode without any iterative process. Schmitz [260] proposed the combination of techniques for online pre-processing and post-processing for an increased MRR in HSM.

The DSST procedure is easy to implement, but it has important limitations. It is useful in the high speed zone (zone C) dominated by clear modes. If the dynamics are complex with many modes limiting the stability, convergence problems with oscillatory solutions may arise [41].

Finally, Zaeh and Roesch [345] introduced a variation in this technique to suppress chatter in robotic milling by locating the process in the ultra-high speed zone (zone D). They propose a minimum spindle speed to reach this area depending on the number of flutes and the measured chatter frequency.

7.2. Continuous spindle speed variation (CSSV)

In the seventies, some authors [130,137,148,272,297] proposed the distortion of the regenerative effect by a Continuous Spindle Speed Variation (CSSV) around a nominal speed. CSSV creates a time dependency in the delay that can improve stability. One of the strong points of CSSV is the flexibility compared to non-uniform tools with irregular flutes described in Section 4, because the parameters of CSSV can be varied easily. The effectiveness of this technique has been demonstrated in several studies in the literature [15,147] (Fig. 27).

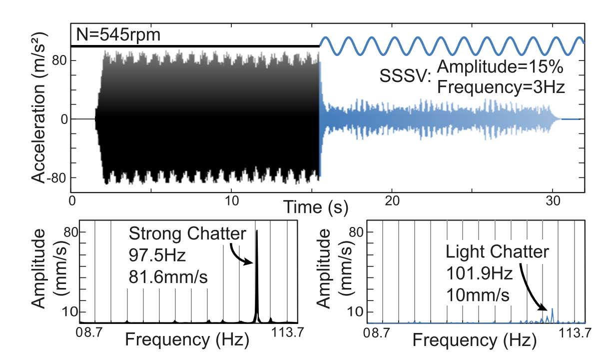


Fig. 27. Effect of continuous sinusoidal spindle speed variation (SSSV) [347].

This technique is based on the introduction of a perturbation in the spindle speed command. There are different methods to vary the spindle speed including sinusoidal [42,250,256], triangular [265], rectangular [272], random [341] or continuous linear perturbation [26]. The sinusoidal signal has emerged as the most efficient technique compared to random, rectangular or triangular shapes. Therefore, the most studied signal is the sinusoidal variation (SSSV-Sinusoidal Spindle Speed Variation), when a simple harmonic variation is applied on spindle speed around the nominal speed value.

The amplitude and the frequency of the variation of the spindle speed should be set up. These parameters are constrained by the tower and spindle limits and the dynamics of the spindle [347]. The variation should not reduce drastically the life expectancy of the different parts of the spindle.

The determination of the optimal amplitude and frequency of CSSV is a complex problem. Only Al Regib [250] proposed a simple formula to select the amplitude and frequency of the variations depending on the chatter frequency and the spindle speed. However, most of the authors used complex stability simulations in which the amplitude and the frequency are varied to find the optimal parameters.

The simulation of the stability of a CSSV process has some additional difficulties. The variable speed transforms the chatter frequency component into an infinite number of frequency components with sidebands at the variation frequency [137].

Inamura and Sata [137] provided a simple function to study the cutting stability with variable spindle speed. However, they found that improvements predicted by the model in stability did not correspond exactly with the experimental results. Sexton and Stone [272] found that results with CSSV are modest, showing their disagreement with some previous researches who claimed important improvements.

The simulations for CSSV optimization in turning and milling operations can be performed by means of initial value time domain simulations [15], multi-frequency methods [256,347], and time domain based methods like semi-discretization [143,347]. In principle, time domain based models are the best options to optimize CSSV, and create a graph were the stability is defined for different amplitude and frequency parameters. These graphs show that the effect of the amplitude is more significant than the frequency variation [347].

The positive effect of the CSSV is circumscribed to the low spindle speed ranges (zones A & B in Fig. 28). In these situations, small variations in the spindle speed can create large variations in the delay between successive waves. Therefore, spindle speed amplitude variations lower than 20% can increase the stability in these zones without any relevant secondary effects. Out of these zones, the SSV is not effective because the variation requirements are not physically reachable.

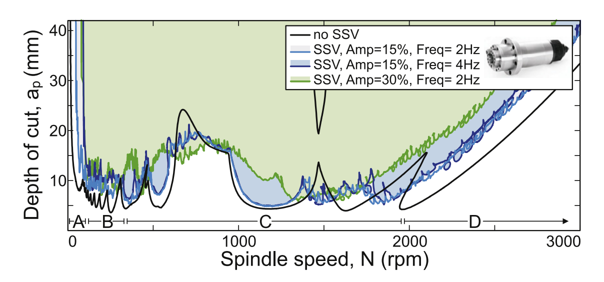


Fig. 28. Effect of SSV in the stability.

These low frequency zones are really complex for ordinary stability simulations because it is important to model the process damping and because the ratio of the critical natural frequencies and the tooth passing frequency is high. Therefore, it leads to large matrices independently of the used numerical method. If the multi-frequency method is selected, an accurate prediction requires a truncation with a high number of harmonics. In time domain based methods like semi-discretization, a dense discretization is necessary for the cutting zone. Consequently, the advantages of these two methods with respect to the complete initial value time domain simulations are reduced.

7.3. Strategies with multiple spindles

Strong chatter problems can arise in parallel milling processes where two or more heads are acting on the same workpiece or the same machine (Fig. 29). The regenerative effect depends on the phase between subsequent tooth passes, and it can be enhanced or reduced by the relation of the different spindle speeds. Thus, the regenerative effect can be perturbed by rotating the two spindles at different speeds. In this case, the regeneration is ruled by two or more constants and different delays. This mechanism is similar to that of non-uniform pitch cutters, but the present method is more advantageous because the speed difference can be adjusted easily by spindle controllers and adapted to the chatter frequency [269].

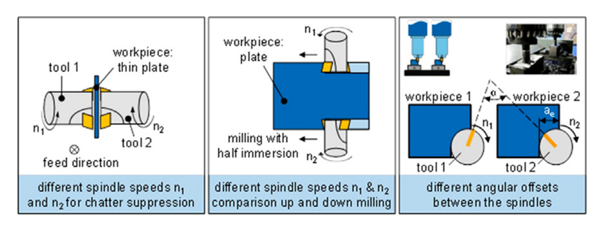


Fig. 29. Chatter suppression for parallel milling processes.

This technique was successfully tested in a double-sided milling process when a flexible part was machined with independently controlled spindles [212,269]. Shamoto et al. [269] proposed an easy tuning formula to establish the difference in the spindle speed ΔN of double sided spindles based on the determination of the chatter frequency f_c:

$$\Delta N = \frac{(2k+1)}{2f_c}. \tag{4}$$

Brecher et al. [56] discussed different strategies to suppress chatter in parallel milling processes on different workpieces, such as the tuning of different angular offsets of the two spindles and the proper selection of different spindle speeds.

Using an analytical expression, Budak et al. [67] demonstrated that the stable material removal rate in parallel milling can be increased substantially by selecting the spindle speeds of both tools properly even for very flexible workpieces.

7.4. Applications

The discrete variation of the spindle speed is an adequate procedure to suppress chatter in the high speed zone with clear dominant modes. In this situation, spindle speed changes can lead to important improvements in stability (zone C). The chatter avoidance in HSM operations of monolithic parts is one of the typical applications of this technique. For instance, Leigh at al. [176] presented results applying this technique to the machining of aluminium jaws of helicopter rotors in industrial environment. The results indicated a fourfold reduction in machining time.

The application of this technique is simple and requires only the measurement of the chatter frequency. Therefore, it can always be used as the first solution to try to reach a more stable point. Portable equipment for the detection and regulation of the rotational speed for chatter suppression exist in the market (Harmonizer® from MLI, BestSpeed® system from KENNAMETAL). The portable system records the vibration by means of an accelerometer or microphone. Nowadays, this algorithm can be found on-line or integrated in the machine.

On the other hand, the CSSV really affects the low frequency region (zone A & B). In these zones, a small variation in spindle speed, lower than 20%, can create a large variation in the delay. This effect was used by Altintas and Chan [15] to design an automatic system to detect chatter and suppress it based on the spindle speed modulation. This low spindle speed zone is relevant in industry, because many chatter problems are restricted to this zone. In fact, the majority of chatter problems in turning operations are good cases to verify the efficiency of CSSV. It is really useful also to improve the stability in hard turning and boring operations.

In milling, the CSSV can be used in some special situations where the tooth passing frequency is low compared to the frequency of the critical mode [42]. This is the case of many milling operations with low machinability materials or special boring operations in heavy duty machining [347]. Lin [179] applied CSSV to drill difficult-to-cut materials such as stainless steel.

The slow regeneration mechanism of grinding processes is also a good situation for CSSV. Moreover, various experimental studies have demonstrated the efficiency of variable grinding wheel speed in cylindrical [161] and centreless grinding [26,37].

CSSV technique has some limitations too. If the damping of the critical mode is low and the process is strongly unstable, the reduction in the vibratory level is not enough to avoid chatter marks. CSSV introduces a special pattern in the surface roughness caused by the rotational speed variation. In some accurate finishing processes, these adverse effects on the surface quality are not acceptable [138].

The DSST and CSSV are effective in different zones of the stability diagram. Bediaga et al. [41] proposed an integrated system for chatter suppression based on the determination of the chatter frequency and lobe order k. Depending on the lobe order, CSSV is activated in high order lobes and DSST in low order lobes.

Several machine tool builders have integrated smart spindle speed strategies for chatter suppression. OKUMA has developed the Navy system in which the DSST and CSSV are used to suppress chatter in turning and milling operations. MAKINO also integrated DSST techniques, and HAAS has also introduced the CSSV in their CNC. MAL Inc. offers a combined DSST and CSSV function as a plug-in module to machine tool builders. The popularity of these techniques has grown in the last years.

7.5. Research challenges

For DSST, the search for the optimal spindle speed in situations with complex dynamics is not trivial. The avoidance of oscillatory solutions requires an improved algorithm.

The CSSV is especially interesting to suppress chatter around the process damping zone (zone A & B). However, the simulation tools developed to define the best variation parameters do not involve this phenomenon. The process damping affects the performance of the CSSV significantly, and must be considered.

The optimal tuning of the CSSV and parallel milling are still an open problem to research. The discovery of simple analytical formulas can help in the automation of solutions. This need is especially true for parallel and multiple milling operations with constant or variable spindle speeds.

8. Active damping techniques

Active techniques are based on the measurement of a parameter related to the vibration, its treatment and then, the introduction of a controlled force signal in response to the measured signal by means of an actuator. This way, a dynamically correlated external energy is applied to the vibrating structure [242].

The most used technologies in active systems are piezoelectric and electromagnetic actuators which proved to be robust [221, 278]. Other smart materials (electrorestrictive and magnetostrictive) and fluids (ER & MR) have been used as well. Smart fluids are basically used in semi active devices, which are essentially passive devices where properties can be tuned in real time, but they cannot introduce directly a controlled force [242].

A classification of active systems can be made depending on how they are positioned in the force flow:

- *Series applications*

  The actuator is located inside of the force path of the machine and has to withstand cutting forces. Therefore, high stiffness is required for these elements. Any breakage of the series actuator results in a machine stop.
- *Parallel applications*

  The actuator is located out of the force path. Hence, the static stiffness of the original structure is maintained. In order to obtain an efficient actuation, the force has to be applied at a location where the targeted mode has large displacement amplitude. The available space is a critical element for parallel applications and the force density should be optimized.

  The rest of this section will review the active chatter suppression techniques based on four main applications: active structural chatter suppression, active tools, active spindle systems and active workpiece holders.

8.1. Active structural chatter suppression

The structural chatter is characterized by the low frequency vibrations, typically between 20 Hz and 200 Hz, of large components of the machine. The use of active techniques is justified because they can cope with the variations of the machine dynamics according to the varying cutting position.

Proof-mass dampers, also known as inertial actuators, use the inertial force created by the acceleration of a suspended mass that is coupled in parallel to the main structure. This active system proposed by Cowley and Boyle [85] is able to damp several modes simultaneously [45,183]. Proof-mass dampers realized with electromagnetic technologies [84] have been used to increase the stability limit by means of a Direct Velocity Feedback (DVF) that increases the damping of structural modes [38]. Munoa et al. [216] used a powerful biaxial electromagnetic damper to suppress structural chatter of a heavy duty milling machine. Proof-mass dampers can also be realized using an electrohydraulic actuator [55], which can generate a large force with a compact design.

Proof-mass dampers are acting in parallel to the force flow but structural modes can also be damped through series applications using piezoelectric actuators or the machine's own drives. Garitaonandia et al. [120] integrated a piezoelectric actuator in an active ballscrew nut to suppress chatter in a centreless grinding machine. One of the easiest methods to actuate in series on a machine tool is by means of conventional drives [24]. Alter and Tsao [11] investigated the use of an actively controlled linear motor to increase turning process stability. Chen and Tlusty [81] simulated the use of an additional acceleration feedback loop to improve the machine dynamics. Kakinuma et al. [151] introduced an active damping system in desktop-sized turning machine, and presented a band-limited force control to suppress chatter in a turning test bench. Munoa et al. [213] increased significantly the stability limit in heavy duty milling by using a rack and pinion drive with an additional acceleration feedback (Fig. 30).

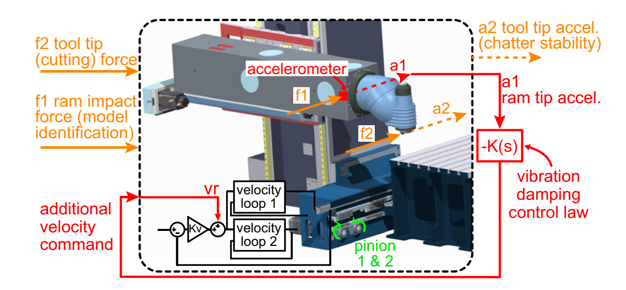


Fig. 30. Active chatter suppression using machine drives [213].

New actuators can be introduced in the machine to introduce force between two points of the structure. For instance, piezoelectric actuators were used by Brecher et al. [53] to compensate the bending modes of the ram of a portal machine. This solution can reduce both the static and dynamic deformation of the structure. Ast el al. [32] inserted a powerful piezo-stack actuator in one of the connecting rods of a parallel machine.

ER fluid dampers are rarely used, but Aoyama and Inasaki [28] introduced them between the table and the frame of a milling machine to change the dynamic compliance.

8.2. Active tools

Boring bars have focused a lot of attention due to their inherent flexibility. Piezoelectric actuators were largely used for this application. Tanaka et al. [299] introduced piezoelectric actuators in the body of the boring bar and applied DVF to increase the damping of the bar. Tewani et al. [301,302] also used a piezo actuator but parallel to the force flow. The vibrations are reduced by a proof-mass absorber generating an inertial force that counteracts the disturbance acting in the system. Pratt and Nayfeh [241] developed a nonlinear model to analyse the active control of cantilever boring bars with a magnetostrictive actuator.

Electromagnetic actuators acting in two radial directions and in the torsional direction were used by Chen et al. [79,184] to increase the damping and dynamic stiffness of a boring bar.

MR fluid was located around the base of the boring bar to adjust the damping and natural frequency by Mei et al.[197]. Later, they optimized the excitation strategy to vary the stiffness of the bar continuously (Fig. 31) [198,340]. Wang and Fei [330] chose an ER fluid instead, and adjusted the dynamics of the bar according to the measured vibrations [329].

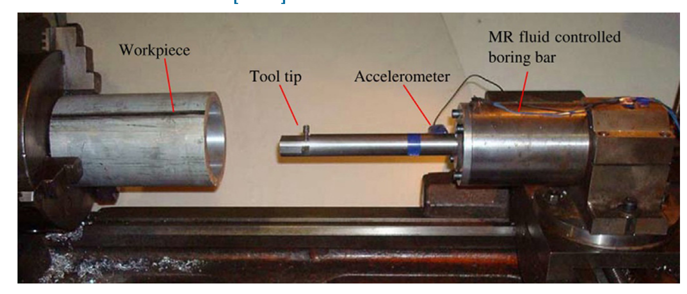


Fig. 31. Boring bar with magnetorheological (MR) fluid [198].

As opposed to the piezoelectric stacks which are embedded in the body of the bar, the other technologies occupy an important volume at the base of the bar.

Other active *turning tools* were also designed to suppress chatter. Lee et al. [174] equipped a turning tool with a proof-mass damper actuated by a piezo element to damp high tool frequency modes at 2000 Hz. Harms et al. [127] proposed an active piezo tool adaptor controlled by an integral force feedback to increase the damping. Weinert and Kersting [335] developed an adaptronic tool holder for deep drilling applications.

For *milling applications*, the possibilities to actively act on the rotating tool are limited. Instead of acting directly on the tool, the research community developed active spindles acting on the spindle's bearings.

8.3. Active spindle systems

Active spindle systems have been developed to suppress chatter coming from the spindle itself or from the flexible milling tool [1]. Two main groups of solutions can be distinguished. First, spindles with Active Magnetic Bearings (AMB) can be applied to increase process stability. Second, active systems able to introduce a force on the spindle axle have been developed.

AMBs are non-contact bearings which support the spindle axle using magnetic forces. AMBs are used in prototypes designed for HSM to reach high spindle speeds. The AMB spindles present advantages in terms of wear and speed compared to traditional ball-bearing spindles. However, the loss of damping is a drawback for process stability which should be compensated through machine control. Knospe [162] used a simplified test bench to evaluate the ability of the magnetic bearing to increase process stability. Van Dijk et al. [95] proposed a robust controller able to increase the depth of cut in a specified spindle range. Huang et al. [131] also simulated the increase of stability that could be obtained by AMBs, but it was not verified in real cutting tests. Gourc et al. [123] modelled the milling stability with active magnetic bearings and provided experimental validations. They emphasized that strong forced vibrations can also limit the depth of cut and integrated this limitation in the stability diagram. Uriarte et al. [320] developed a magnetically levitated spindle in which they were able to compensate the loss of damping with DVF control that integrates phase compensation.

Active systems can also be placed on the spindle to suppress chatter. Albertelli et al. [10] used piezoelectric actuators able to control the position of the front bearing of the spindle. A similar system is presented by Monnin et al. [209,210] (Fig. 32). As in [95] and [162], two control laws are proposed. On the one hand, a disturbance rejection control law optimizes the tool tip FRF. On the other hand, knowing the process, the controller can optimize the stable depth of cut. Dohner et al. [97] placed the electrostrictive actuator on the spindle tip. Denkena and Gümmer [92] located three piezo actuators and proposed to generate a superposed vibration to disturb the regenerative effect.

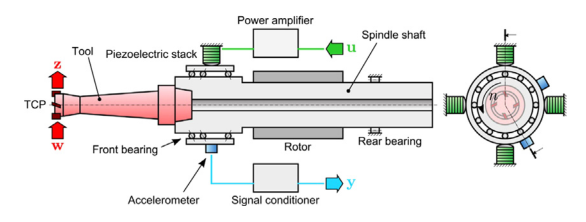


Fig. 32. Active spindle [209].

8.4. Active workpiece holders

Active workpiece holders can be used to suppress chatter by influencing the relative movement between the cutting tool and the workpiece [35]. Due to the high dynamics and positioning accuracy in the dynamic load range, piezoelectric actuators are best suited for this type of applications. Rashid and Nicolescu [247] validated the feasibility of such systems. Abele et al. [4] designed the workpiece holder with FEM taking into account the piezo actuators with high bandwidth and low force constraints. Brecher et al. [54] developed a workpiece holder to suppress chatter coming from the low frequency structural flexibilities of the machine (Fig. 33). Indeed, active workpiece holders can be used to damp vibrations generated by other elements of the machine. Zhang and Sims [355] proposed to damp a flexible workpiece using a piezoelectric patch. Parus et al. [237] suppressed chatter coming from a flexible clamping system.

8.5. Active control algorithms

All the previously cited articles of this section use control algorithms to operate their systems. The typical stability,

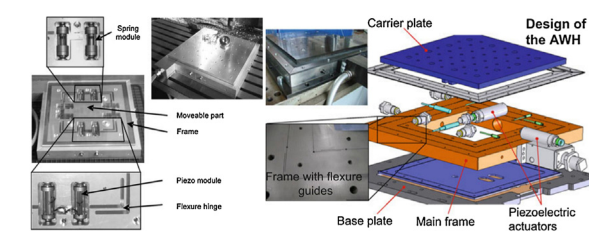


Fig. 33. Active workpiece holder [4,54].

performance and robustness issues have to be addressed. Preumont [242] presented relatively simple but efficient control strategies which have proved to be useful. He favoured collocated actuator/sensor pairs with simple controls such as DVF when the actuator is producing a force (proof-mass dampers) and integral force feedback when the actuator is producing a displacement (piezo stacks). Ehmann and Nordmann [104] compared those different active control configurations on a given structure and concluded that proof-mass dampers located close to the tool tip can achieve better performances.

The DVF suppresses chatter at the lower stability area. However, with clear dynamics, it does not significantly change the stability limit on the sweet spots of the high speed zone (zone C). A delayed acceleration feedback has been proposed to improve the stability far from the stability minimum [186,216].

Huyanan and Sims [132] proposed a virtual passive absorber which focuses the effect of the actuator close to the problematic mode. However, a model of the targeted mode is needed whereas the DVF control does not use any model. Vetiska and Hadas [327] developed a simulation environment with multi-body dynamics simulation and a control design.

To face practical implementation issues, Hardware In the Loop (HIL) simulators allow to study the active damping strategies for chatter suppression [4,118,132,185]. Root locus analysis of chatter was performed by Ganguli et al. [119] using Padé approximation to check the stability of the control law.

Apart from the simple and efficient control strategies, authors have looked for optimal and robust controls using μ-synthesis [95] or state feedback [271]. Zhang et al. [354] took into account the actuator power limitation with a model predictive control. The delay of the controller is usually problematic and several strategies have been proposed to compensate based on a lead compensator [185,320].

Finally, Olgac and Hosek [222] proposed to use delayed resonator to suppress chatter, and Stepan and Insperger [288] introduced the act and wait control method to control time-delayed systems.

8.6. Applications

The use of active techniques generally implies a laborious set-up (actuator, sensor, power supply etc.) in a hostile environment. These economical and technical efforts are justified only when other simpler solutions cannot provide an optimum response. This is the case when the dynamics are changing, when multiple modes have to be damped or when space constraints prevent the use of passive dampers.

For structural chatter, creating weak systems by introducing an active component in the force flow seems to be too risky. A possible solution is to use the machine tool drives which can suppress chatter with low implementation effort as it uses the actuators, sensors and controller already installed in the machine. CNC providers are working in this field, and Heidenhain offers an Active Chatter Control [128,155]. However, for structural chatter suppression, the use of a proof-mass damper close to the tool tip with collocated control offers better results. The electromagnetic technology proves its robustness and has been implemented commercially (Fig. 34).

Boring bars attracted lot of publications. However, it seems hard to surpass the passive dampers in standard conditions. Thus,

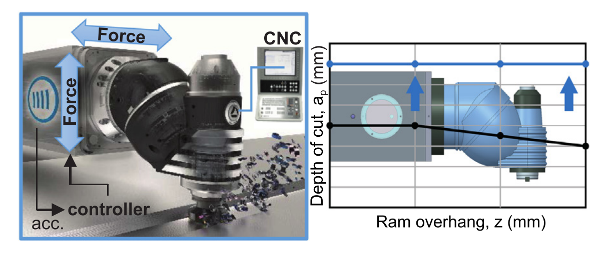


Fig. 34. Dynamic Active Stabilizer (DAS®) by SORALUCE.

extreme situations with slenderness reaching twenty times the diameter and with large diameters allowing the use of active proof-mass dampers at the tip could be a potential application. Piezo actuators integrated in the holder of the bar could also be considered as the space requirement is really small. The possibility of varying the bar overhang and thus the bar frequency is also a key element.

8.7. Research challenges

Three main research challenges are encountered in this paragraph.

First, it is still difficult to size the actuator accurately for a given application. Only few works are coupling the cutting process models and actuator models to analyse the constraints in terms of force capacity, actuation and measurement point's location etc. [187].

Suppressing high frequency chatter resulting from workpiece flexibility is still a challenge. The variation of the dynamics with the material removal increases the complexity of this problem.

Thirdly, almost none of the prototypes presented before have been industrialized [221,235]. Indeed, industry needs efficient and reliable solutions providing a clear breakthrough to justify the cumbersome implementation linked to active systems.

These challenges require the determination and the design of the optimum design strategy to create a robust active system.

9. Conclusions

The following procedure can be followed to evaluate the best solution to tackle a chatter problem. First, confirm that the vibration problem is caused by chatter [244]. Second, measure the chatter frequency. Third, evaluate the position in the stability diagram (order of the lobe k). Fourth, measure the dynamic stiffness/receptance to identify the critical mode(s). Fifth, obtain the shape of the critical mode by means of a modal analysis. This rapid evaluation gives crucial indications for the selection of a proper chatter suppression technique (Table 6).

In zone A of process damping, the CSSV is always a realistic troubleshooting solution. Special tool geometries with special edges or variable helix can also increase the process damping effect. Finally, if the implementation of the previously mentioned solutions is not enough, stiffness and damping increase are effective if there is space (boring bars, fixture or machine structure). The effect of the cutting edge geometry and the wear is important in this zone.

In zone B, variable helix and pitch tools could be a good option, but they require an optimization of the geometry. The CSSV could be envisaged but the required oscillation amplitude might be too large. Here again, stiffness and damping improvements are effective to be able to increase the minimum stability limit.

In zone C, the SLD help the process planner to find the sweet spots. This situation often occurs in HSM of aluminium alloys with flexibilities coming from the tool or spindle. When the process parameters cannot be changed due to machinability limitations, variable pitch tools are a good solution. If the process parameters and tool are frozen or the process variations are not feasible, the stiffness or damping improvements techniques should be envisaged. These last solutions are also appropriate when the chatter is produced in universal machine tools where many different machining operations can be programed and some of these operations are highly limited by structural chatter

In zone D, the higher the spindle speed is the more improved the stability is. However, this area is generally not reached because of spindle limitations or because of other chatter coming from higher frequency modes that start limiting the productivity.

Chatter occurs more often in roughing operations due to a larger chip load and the depth of cut. The use of serrated cutter and high feed inserts can be a good alternative for chatter-free roughing operations independently of the zone.

At the present, the active solutions can be considered to solve structural chatter problems in heavy duty operations with low chatter frequency. Machine drives can also be used to maximize the damping of the structural low frequency modes.

To conclude, even if the research community should still improve the robustness and simplicity of the use of many chatter suppression techniques, the production engineers already have a wide range of solutions to fix industrial problems.

Table 6
Application of chatter suppression techniques.

Origin	Zones in the stability diagram			
	Zone A	Zone B	Zone C	Zone D
Tool/boring bar	CSSV Passive damper Variable helix	Passive damper CSSV Variable helix/pitch.	DSST SLD Passive damper Variable pitch	increase spindle speed
Spindle	CSSV Variable helix. Edge geometry	CSSV Variable helix/pitch	DSST SLD Variable pitch	increase spindle speed
Workpiece	CSSV Variable helix Increase stiffness	CSSV Variable helix Stiffness increase	DSST Increase stiffness	increase spindle speed
Fixture	CSSV Variable helix Passive damper	Passive damper Stiffness increase	SLD process passive damper Stiffness increase	increase spindle speed
Machine structure	CSSV Passive damper	Passive damper CSSV Stiffness increase Active damping	Passive damper Stiffness increase Active damping	increase spindle speed

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